

CONTINUOUS CENTRIFUGING

OF SHEEPSKINS

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SYMBOLS USED IN THE TEXT

(Where a written description is inadequate the page number of an appropriate figure is given).

a	radial spacing between adjacent spiral turns
A	constant
B	constant
C_D	drag coefficient
d	fibre diameter
e	eccentricity, p.69
f_a	alternating component of stress
f_m	constant component of stress
g	acceleration field magnitude, p.10, 32 ft/sec ² all other pages
h,H	height (axial dimension)
h_o	fibre height in direction of acceleration field
$k_{\text{subscript}}$	constant
L	machine geometry dependent length, p.69
m	mass
n	number of turns in the spiral
P	power
r	radius
r_i	radius of the inner end of the spiral
r_{in}	radius of the entry hole
R	machine radius (rotating parts)
R_o	radius of the outer end of the spiral
t	time
t_{min}	minimum hold time
t_{mean}	mean hold time
t_{max}	maximum hold time
v	velocity

V	velocity at maximum machine radius
α	centrifugal acceleration magnitude
β	machine geometry angle, p.75
γ	surface tension
θ	exit angle when referring to paddle-gated machine, p.49; machine geometry angle when referring to eccentric spiral machine, p.68.
θ_b	contact angle
ω	angular velocity
ω_1	mainshaft angular velocity
ω_2	drum angular velocity relative to crank
μ	coefficient of friction
ρ	fibre density
σ	stress
ϕ	machine geometry angle, p.69
ψ	spiral geometry angle, p.69.

HISTORICAL BACKGROUND

Freezing companies in New Zealand have long been aware of the desirability of a continuous automatic system for centrifugally reducing the moisture content of woolly sheepskins after washing. The problem of designing such a machine was brought to the attention of the Department of Mechanical Engineering, University of Canterbury in 1969 and was used as the subject of a design study for final year Bachelor of Engineering (Mechanical) students in 1970.

A variety of systems were proposed, most being based on Rands⁽¹⁾ idea of a rotating drum lined with conveyor belts. The design offered by the author used bladed wheels to control the motion of the sheepskin through the machine. It was considered that this principle of operation could be the basis of a successful machine. A grant of \$15,000 (NZ) was made by the New Zealand Freezing Companies' Associations to the University to finance the design, construction and development of full-size machinery for continuously centrifuging woolly sheepskins.

SUMMARY

One stage in the processing of sheep and lamb skins in New Zealand freezing works is the removal of water from the woolly skin after washing.

Neither of the two systems used at present to achieve this reduction in moisture content is entirely satisfactory. Passing the skin between pressure rollers often results in damage to the skin or wool. Batch centrifuging in manually loaded and unloaded "hydros" (basket centrifuges) involves high labour and capital costs.

Initial investigation into alternative methods of water removal indicated that an automatic continuous centrifuging system could be economically preferable to the existing processes. This thesis describes the design and development of machines constructed with the ultimate aim of producing a commercial continuous centrifuge capable of automatically effecting the required water removal.

Two full size machines were designed and constructed. The second machine was developed to the stage where reliable skin control was achieved, and the principle of operation could form the basis for the design of a successful commercial machine.

The effect on wool moisture content of variations in centrifuging parameters was investigated.

CHAPTER ONE

INTRODUCTION

By far the greater part of New Zealand's export earnings result from the sale of primary products. The processing of these products to the stage where they are useful to overseas markets is an important part of New Zealand industry.

Some thirty five million sheep and lambs are killed every year in New Zealand freezing works, so this is an industry in which small changes in unit processing costs has a significant effect on profitability. Increasing competition from synthetic materials necessitates the streamlining of processing in freezing works. This thesis describes investigations into increasing the efficiency of one operation in woolly sheepskin processing prior to the sale of wool and pelt.

Following slaughter, the sheep or lamb is skinned. Both the skin and the wool it carries are commercially useful products. The woolly skin is then transported to the felmongery section of the freezing works while the carcass continues along a separate processing line.

The skin is washed in cold water, often in automatic continuously operating spray washers. This cleans the wool and skin to some extent, and cools it, thus delaying bacterial action. The skin remains in the wash for sufficient time for the wool to become saturated and for the skin to absorb a significant quantity of water. Reduction in the moisture content of the woolly skin to a level suitable for subsequent processing is then effected, either by

passing the woolly skin between closely spaced "squeeze rollers", or centrifuging in manually loaded and unloaded "hydros".

Squeeze rolling has the advantages of being a continuous process, but is incapable of reducing the wool moisture content to the levels obtained by centrifuging without extensive damage to both the skin and wool. Due to the inability of the roller system to allow for variations within or between the skins, and incorrect feeding of the skin into the rollers, the method results in a high incidence of grain damage to the pelt.

Squeeze rolling systems have been replaced in many freezing works by banks of manually loaded and unloaded "hydros". Although batch centrifuging requires more manual work than squeeze rolling, it is often considered worthwhile as it gives better results. Centrifuged wools have a better colour and handle better than squeeze rolled wools, and wool tips are not as brittle after drying. The lower wool moisture content of centrifuged skins reduces energy requirements for hot air drying of the wool to marketing moisture level.

After water extraction the skins are treated on the flesh side with a lime-sulphide depilatory paint which penetrates the skin and attacks the wool follicles, allowing the wool fibres to be pulled from the skin. Better penetration and consequently easier removal of wool occurs with higher moisture content. The removed wool is then sorted and further dried by continuously operating hot air dryers to the level required for marketing. The skin is further processed before sale for use as shoe and garment leather.

The "hydros" used for extracting water from sheepskins are typically five to six feet in diameter with an operating speed in the range 700-800 r.p.m. A freezing works equipped for a peak killing rate of 1600 lambs per hour would require six "hydros" with one operative per machine. The cost of each "hydro" is approximately \$NZ 14,000. Floor space required for these six machines is approximately three hundred square feet. The total average power consumption at peak killing time is approximately 20 KVA. Each machine has four starts per hour, load, run and unload times being each about five minutes for each machine cycle.

An automatic continuous system for centrifuging woolly skins could have several advantages over batch centrifuges. Economically the most important advantage would be a saving in labour costs which was estimated⁽¹⁾ in 1968 to be approximately \$NZ 130,000 per annum. If a continuous machine could replace a number of batch machines, a saving in capital cost and floor space could result. Power consumption of a continuous machine could be lower than for its equivalent capacity in "hydros", as no energy would be lost in accelerating machinery after the initial start, and the motor would always be running at design speed.

Existing systems used for continuous centrifugal separation of other materials were examined and were considered unsuitable for sheepskins. Other industries, for example, commercial laundries, where water is centrifugally removed from garments similar in physical characteristics to sheepskins, use batch centrifuging systems.

Reduction in moisture content of the woolly skin by means other than centrifuging was considered. Heated air

drying is impractical for several reasons; the differential drying that would occur would result in over-dried skin and wool tips; elevation of temperature of the skin would increase bacterial action; energy requirements would be very high, and cost in excess of \$NZ1m for the total annual kill.

Linear and vibratory acceleration systems were briefly investigated, but were found to be impractical because of size or stress limitations.

The economic advantage which a continuous centrifuge for sheepskins could offer is more significant in New Zealand than in other major wool producing countries. The sheep population of New Zealand (56.7m, 1973⁽⁸⁾) is less than that of two other countries, U.S.S.R. (142.8m, 1973⁽⁸⁾) and Australia (140.1m, 1973⁽⁸⁾). Labour costs in New Zealand are higher than in U.S.S.R. Australian merino slipe wool is much more valuable than the merino pelts, and the process used to separate wool and skin is designed to give optimum wool yield with some sacrifice in pelt quality. This process ("sweating") does not require reduction in wool moisture to the levels desirable for the New Zealand process, and squeeze rolling is adequate.

I.1 OBJECTIVES OF THE PROJECT

The aim of this work was to develop a process for reducing the moisture content of woolly sheepskins which showed an economic advantage over existing systems. The objectives of the work were then redefined more specifically as :

1. To conceive, design and construct a machine suitable for the mechanical handling of sheepskins in a rotating system; and
2. To adjust process parameters (hold time and acceleration

field) to achieve the required degree of water removal.

With these objectives established, investigations into the many phenomena associated with the process were continued only as far as their results made a contribution towards achieving these aims.

I.2 CHRONOLOGICAL OUTLINE OF THE WORK

Table 1 (p. 8) shows the sequence of investigations culminating in the final design.

Although not apparent from this table, at least one third of the time devoted to the work as a whole by the author was spent in manipulation of ideas for possible principles of operation for a continuous centrifuge. While the majority of this time was not immediately productive, there would appear to be no alternative when an entirely new system is necessary. The idea of the eccentric spiral system arose eventually not as a logical extension of any existing systems, but as a consequence of this time devoted to manipulation of ideas. Any attempt to describe the process resulting in the conception of a new idea, such as the eccentric spiral system, or the paddle-gated system, can only be superficial.

I.3 REMOVAL OF WATER FROM WOOLLY SHEEPSKINS

The performance of "hydros" in freezing works had shown that centrifugal methods could give a useful reduction in the moisture content of wool carried on sheepskins. The extent of water removal from a woolly skin subjected to acceleration is affected by :

1. The magnitude of the acceleration field;
2. Hold time in the acceleration field;

TABLE 1CHRONOLOGICAL SEQUENCE

POSSIBLE ADVANTAGES OF A CONTINUOUS CENTRIFUGE RECOGNISED BY
THE N.Z. FREEZING INDUSTRY

RANDS' INVESTIGATION AND PROPOSALS

PROBLEMS OF MACHINE OPERATING PRINCIPLE AND MECHANICAL DESIGN
BROUGHT TO THE ATTENTION OF THE UNIVERSITY

FINAL YEAR DESIGN STUDY (D.V. WESTON & OTHERS) ON CONTINUOUS
CENTRIFUGING OF SHEEPSKINS

AUTHOR'S PROPOSAL (PADDLE-GATED TROUGH SYSTEM) CONSIDERED
WORTH DEVELOPMENT

GRANT MADE TO UNIVERSITY

POSTGRADUATE STUDY COMMENCED

INVESTIGATIONS INTO RETENTION OF WATER IN WOOL, ETC.

INVESTIGATIONS INTO ALTERNATIVE FORMS OF WATER REMOVAL

INVESTIGATIONS INTO EXISTING CONTINUOUS CENTRIFUGING MACHINERY
FOR OTHER MATERIALS

DECISION TO COMMENCE DESIGN AND CONSTRUCTION OF PADDLE-GATED
TROUGH SYSTEM

TESTING OF COMPONENTS AND SUB-ASSEMBLIES - TESTING OF COMPLETED
MACHINE

FAILURE OF PADDLE-GATED TROUGH SYSTEM TO PERFORM THE REQUIRED
MECHANICAL HANDLING

MODIFICATIONS - OF HELP, BUT RESULTS STILL UNSATISFACTORY

RE-EVALUATION OF MECHANICAL HANDLING TECHNIQUES

PERIOD OF EXERCISE OF INVENTIVE/CREATIVE FACULTIES - NO USEFUL
RESULT

CONSTRUCTION OF EXPERIMENTAL FRUSTUM RIG

EXPERIMENTS WITH FRUSTUM MACHINE

PERIOD OF EXERCISE OF INVENTIVE/CREATIVE FACULTIES

ECCENTRIC SPIRAL PROPOSAL

DESIGN AND CONSTRUCTION OF ECCENTRIC SPIRAL MACHINE

MODIFICATIONS TO ECCENTRIC SPIRAL MACHINE

SUCCESS IN MECHANICAL HANDLING OF SHEEPSKINS

MODIFICATIONS TO REDUCE WOOL MOISTURE CONTENT

PERFORMANCE EVALUATION ON ECCENTRIC SPIRAL MACHINE.

3. Physical restrictions on the escape of water, including number and placing of drain holes in the centrifuge, proximity of other skins and the orientation of the skin with respect to the acceleration field;
4. Individual skin and wool characteristics.

I.3.1 Retention of Water in Wool

Water may be retained in a bundle of wool fibres both as intrafibre and interfibre moisture. Intrafibre moisture is retained both by hydrophilic groups in the chemical structure of the wool and by surface tension forces acting on small volumes of water in minute cracks and crevices in the fibres. Interfibre water is held by surface tension forces acting both on small droplets on the surface of a fibre and on small volumes of water between fibres. Below a wool moisture content of approximately 25% all water is preferentially retained as intrafibre moisture within the fibres with no interfibre moisture. In the range of wool moisture contents considered in this thesis (greater than 35%) some water always exists external to the fibres, and the intrafibre hygroscopic sites are saturated.

I.3.2 Effect of Acceleration Field on Equilibrium Wool Moisture Content

When a large number of wet closely packed fibres are centrifuged the spaces between the fibres may be considered as capillary tubes. A theoretical relationship between retained water and centrifuging parameters has been derived⁽²⁾. This

relationship is the result of equating centrifugal force on a capillary to surface tension force. The equilibrium condition approached with increasing time is given by the following relation :

$$R = \frac{\gamma}{g} \cdot \cos\theta_b \cdot \frac{1}{h_o} \cdot \frac{1}{(\rho d^2)^{\frac{1}{2}}} \cdot k$$

Where R = regain (= mass of retained water/mass of dry fibres)

γ = surface tension

g = acceleration field

θ_b = contact angle

h_o = fibre height in direction of acceleration field

ρ = fibre density

d = fibre diameter

k is a constant.

This relationship is obtained from consideration of an ideal case (that of uniform parallel fibres), the mathematical model only approximately representing the physical phenomena even for this ideal case, and therefore, predictions using this relationship may not be sufficiently accurate for practical centrifuging. However, it was considered useful in anticipating trends in the extent of water removal for various centrifuging and wool fibre conditions. It was therefore expected that if a woolly sheepskin were subject to a centrifugal acceleration field for sufficient time, then :

1. The regain of the wool would be lower at higher acceleration field magnitudes.
2. Longer fibres would have lower regain.
3. Larger diameter fibres would have lower regain.

4. The orientation of the fibres would affect the regain.

The concept of centrifugal forces balancing capillary forces led to further conclusions :

- (a) That the most uniform water removal would occur over any one fleece if the skin were subject to the acceleration field for sufficient time for this force equilibrium to be closely approached. If a higher acceleration field for a shorter time were used to achieve the same average wool moisture content, then the wool moisture content in any one area of the fleece would depend largely on the ease of escape of water from that particular region.
- (b) That the acceleration field suggested by Rands⁽¹⁾ (140g) should give satisfactory wool moisture content irrespective of machine characteristics provided sufficient time is allowed for the escape of water.

I.3.3 Effect of Hold Time on Wool Moisture Content

As described in the previous section, it was anticipated that (neglecting loss by evaporation), an equilibrium moisture content would be approached with increasing time, the rate of water loss diminishing with increasing time.

I.3.4 Restrictions on the Escape of Water

Restrictions on the escape of water from the wool were divided into three categories :

1. The resistance to fluid flow through closely packed wool fibres.

2. Retention of water in concave pockets in the skin.
3. Drain holes in the machine wall supporting the skin.

The resistance to the flow of water through the wool fibres is dependent on wool characteristics (fibre diameter, length, greasiness, etc.), the orientation of the fibres relative to the direction of flow, and the extent to which the acceleration field compresses the wool.

Skin orientation in a centrifuge may be such that pockets of water form in the skin. Permeability tests were performed on pieces of shorn skin, first with the flesh and then with the wool side subjected to water pressures of up to 20 p.s.i. It was found that the skin was practically impervious to water, in both directions, only a few drops per minute passing through a 1.5 in² area with a pressure differential of 20 p.s.i.

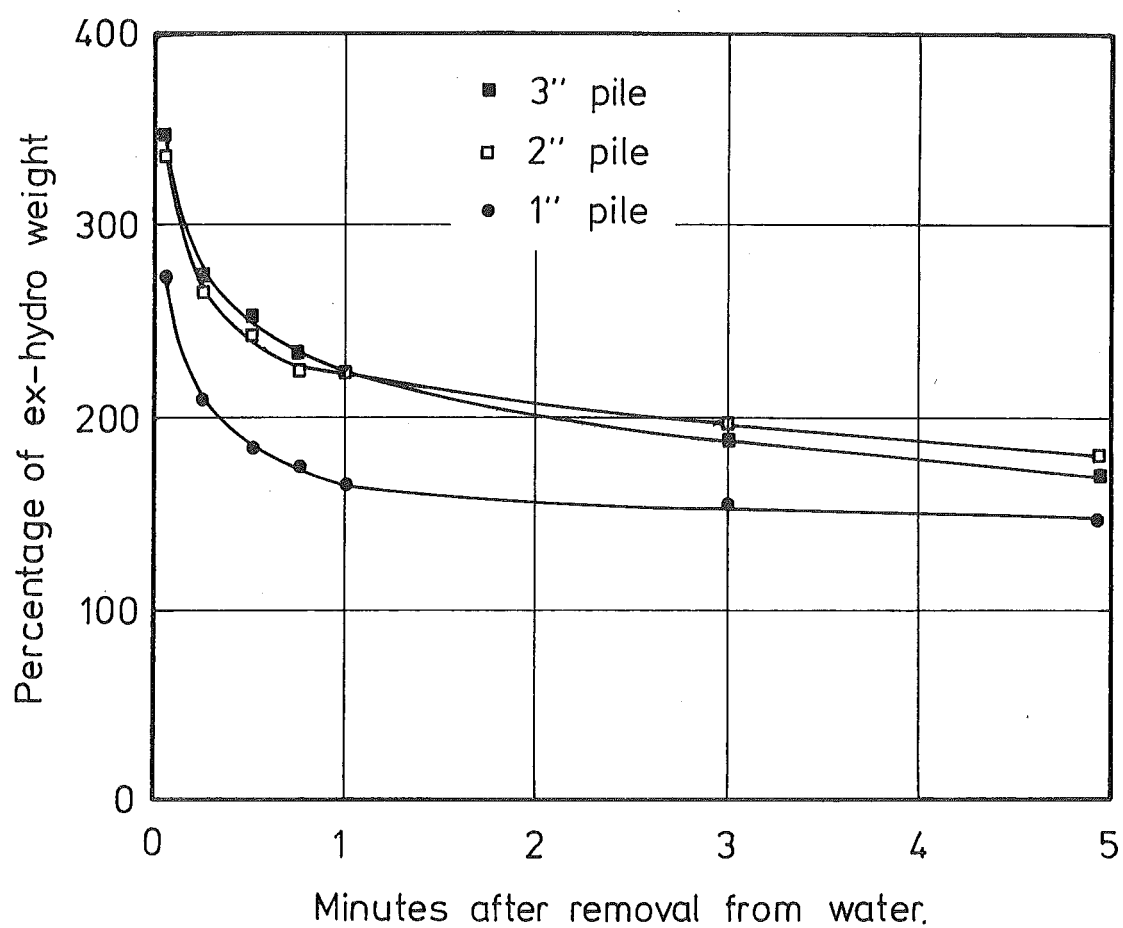
The number and configuration of drain holes was not expected to influence wool moisture content significantly, providing the restriction on water escape imposed by them was less than that from other restrictions on water flow; i.e. for a drain hole capacity above a critical minimum, increase in the number of drain holes could be expected to have little effect.

I.3.5 Unigravitational Draining of Water from Woolly Skins

Experiments were conducted on unigravitational water loss from woolly sheepskins. Three skins were soaked for .5 hr and hung over racks to drain for four hours. To reduce evaporation loss the skins were then enclosed in plastic bags with a small drain hole, and allowed to drain a further ten hours.

Each skin was weighed twelve times during this fourteen hour period. The results are plotted in Figures 1 and 2

FIG. 1. UNIGRAVITATIONAL DRAINING OF THREE WOOLLY SHEEPSKINS OVER A FIVE MINUTE PERIOD.



(pp. 13 & 15). The weights of the skins approached limiting low values with increasing time, somewhat higher than their respective ex-"hydro" weights. This result is compatible with the force equilibrium theory.

The power consumption of a continuous centrifuge increases with increasing mass input rate. It is therefore desirable to reduce the weight of wet skins entering the machine by allowing a unigravitational drain period. Accumulation of a large number of skins on a draining conveyor is undesirable because of the floor space necessary and the lag introduced into the process line.

It was found that half a minute drain time will give a reduction in wet skin weight of approximately 25%. In many freezing works the spacing between centrifuges and the washer is sufficient to allow this drain time on the existing conveyor system by reducing belt speed.

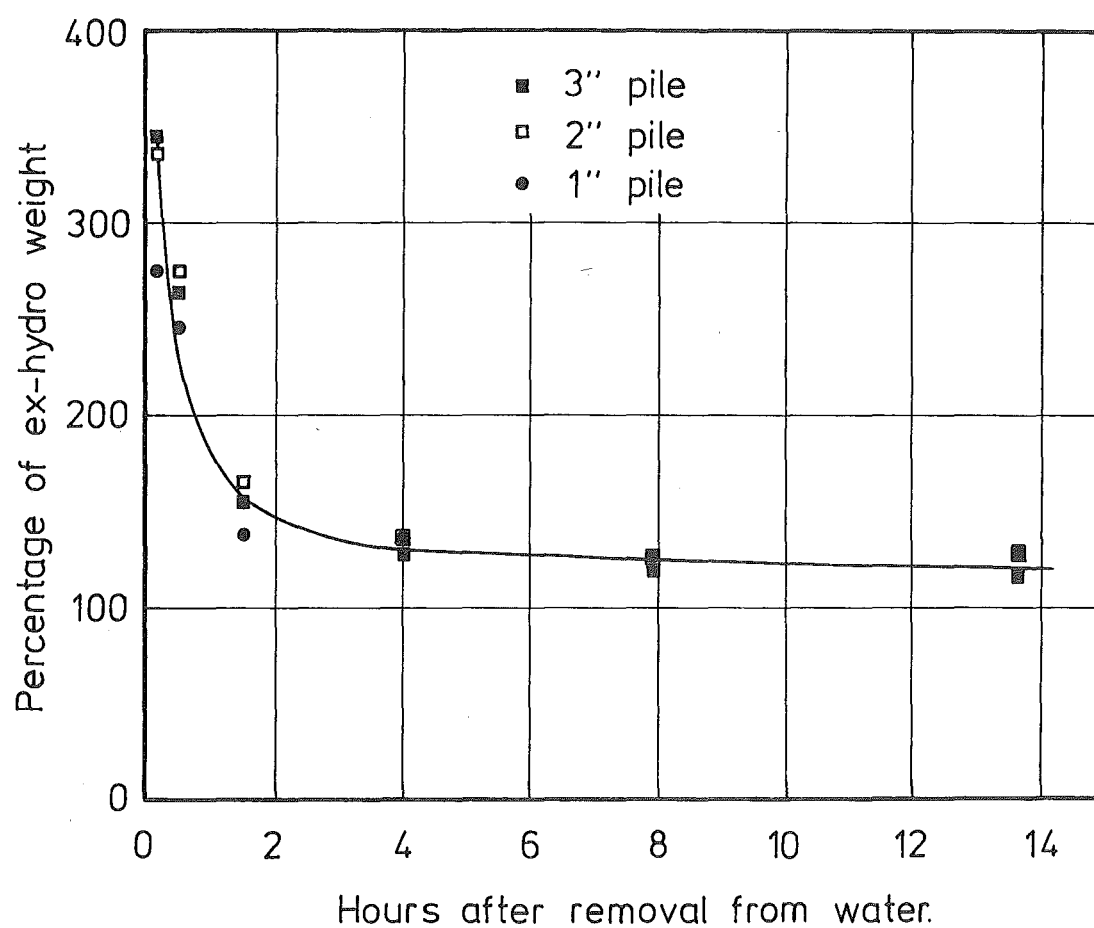
Prior to passing skins through machines described in this thesis, approximately half a minute unigravitational draining was allowed.

I.3.6 Skin Moisture Content

Skin moisture content (as opposed to wool moisture content) after centrifuging also has important effects on subsequent processing. Water absorbed by the skin during the wash is essential for the transport of depilatory chemicals from the flesh side of the skin to the wool roots. Poor depilatory action and consequently difficult pull of the wool may result from low skin moisture content.

Whereas thorough soaking of the wool fibres on the woolly skin occurs in a few minutes in an agitated wash system, the

FIG. 2. UNIGRAVITATIONAL DRAINING OF WOOLLY SHEEPSKINS.



skin continues to absorb water even after twenty minutes washing. The test used during this work to determine whether sufficient water remained in the skin after centrifuging was a practical wool-pull evaluation. All skins returned to freezing works after centrifuging in the machines described in this thesis gave a satisfactory pull after treatment with the same depilatory paint used for conventionally hydro-extracted skins.

I.4 REQUIREMENTS OF A CONTINUOUS MACHINE

A continuous centrifuging system for woolly sheepskins can be justified only if it has economic advantage over the existing banks of batch centrifuges. The total cost of the process in either case is made up of a number of parts, the most significant being :

1. Labour.
2. Capital, depreciation and maintenance.
3. Effect of the process on the cost of subsequent processes.

A continuous centrifuging system could be more costly in one or more of these parts and still offer an overall economic advantage.

When assessing the merits of a proposed system, they were compared both with the hypothetical ideal system and the best existing system. In the following list of requirements of a continuous centrifuge both the "ideal" system and the existing batch system are considered where appropriate :

- (a) The operation of the machine should be such that a sheepskin fed into the machine is held in a

centrifugal acceleration field for a controlled time while allowing water to escape, and then discharged. Inventing a machine which could reliably perform this mechanical handling operation was the most demanding part of the work described in this thesis.

- (b) Ideally the system should damage neither wool nor skin. Damage to the product as a result of "hydro" extraction is unusual.
- (c) Ideally the system should reduce the moisture content of the wool uniformly to that required for marketing (9 - 18% regain), while allowing sufficient water to remain in the skin to give satisfactory depilatory action. "Hydros" reduce wool moisture content to approximately 44% mean, standard deviation over any one skin being typically 3% to 4%. Centrifuging for longer periods in hydros results in impaired depilatory penetration. An average wool moisture content of 45% is considered satisfactory by the industry.
- (d) Throughput capability should be as high as possible. A single conventional hydro has a throughput of approximately 270 skins per hour.
- (e) Ideally no operator should be necessary for the machine. Hydros require the equivalent labour of one man per machine.
- (f) The machine should be suitable for a form of construction similar to other freezing works machinery, i.e. rugged, low carbon steel structural parts, devoid of small tolerance fits, delicate

components, mechanisms requiring accurate or frequent adjustment.

- (g) Control over centrifuging parameters (hold time and/or acceleration field magnitude) is desirable to make allowance for seasonal variations in skin properties.
- (h) The system should ideally be capable of accepting skins arriving at random time intervals.

I.5 CENTRIFUGAL FIELD MAGNITUDE AND HOLD TIME SUITABLE FOR CONTINUOUS CENTRIFUGING

During centrifuging in conventional hydros the skins are subjected to centrifugal acceleration for approximately five minutes. At peak killing rate $3\frac{1}{2}$ minutes of this time may be spent accelerating the centrifuge and its load, and the remainder braking. The machine may not always reach full speed. With normal hydro loading techniques skins are several deep in the radial direction. Water removed from skins closest to the centre of rotation must pass through or around other skins before escaping from the machine. Skin orientation is not generally controlled during loading of skins into hydros.

It was considered that a continuous centrifuging system could achieve a similar degree of water removal in less time than the five minutes necessary for satisfactory hydro extraction. A continuous machine could be designed to accelerate the skins to full rotational speed and decelerate the skins to rest in less than the corresponding times for a hydro. The skins would, therefore, spend less time in an acceleration field magnitude lower than design maximum. A continuous

machine would cause less restriction on escape of water from the skin if the handling mechanism was designed such that skins were processed one deep in the radial direction.

In an attempt to determine a combination of hold time and acceleration field magnitude suitable for continuous centrifuging Rands⁽¹⁾ performed two series of experiments. In the first series small pieces of woolly sheepskin were centrifuged in such a manner that water could escape from the wool without being trapped in concave pockets in the skin. In the second series whole woolly skins were dropped into a running "hydro". After a measured time at that test speed the hydro was brought to rest by application of the brake. Rands concluded from these experiments that "a gravitational field of about 140g for five or more seconds will give wrung wool moistures similar to those obtained by conventional "hydroing".

I.6 VARIATIONS BETWEEN WOOLLY SKINS

Sheepskins vary widely in properties.

A small lambskin with .5 in pile may have a wet (ex-washer) weight of five pounds, and a large skin with 6 in pile may have a wet weight of fifty pounds.

Wool fibre diameter may vary from ten to sixty microns.

Fibre density (number of fibres per unit area of skin), and wool grease and suint content also vary between individual skins and between areas of any one skin.

It is apparent then that a practical continuous centrifuge must be capable of accommodating a wide variance in input parameters.

CHAPTER TWO

MACHINES FOR CONTINUOUS CENTRIFUGING

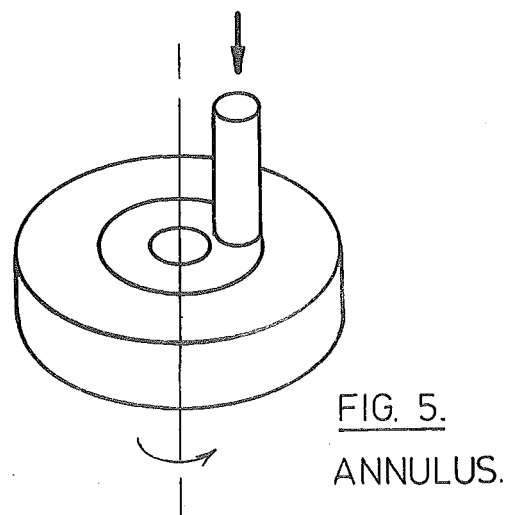
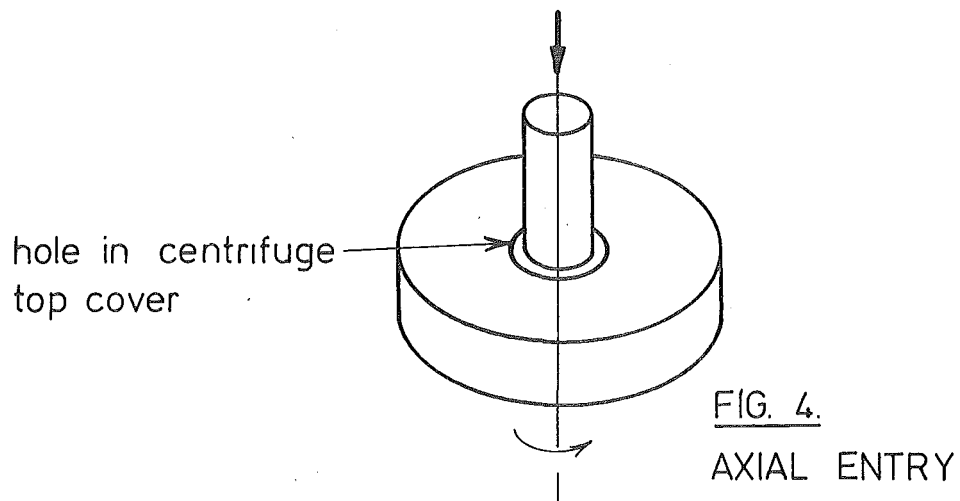
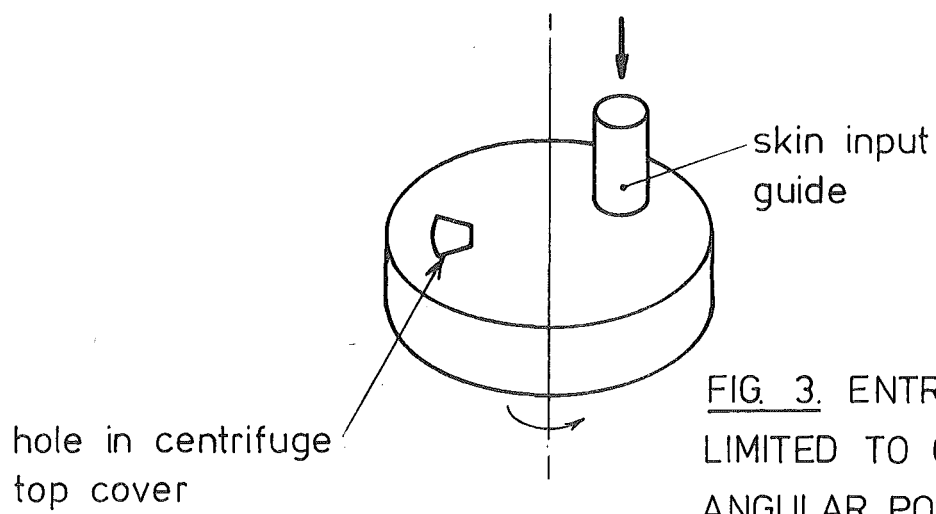
II.1 INITIAL CONSIDERATIONS

The mechanical handling necessary for a continuous centrifuge for sheepskins may be divided into three stages :

1. Introduction of the skin into the rotating system.
2. Holding of the skin in the centrifuge acceleration field for a set time.
3. Removal of the skin from the rotating system.

From the results of Rands' experiments it was expected that an acceleration field in excess of 100g would be necessary to achieve the required degree of water removal. If the skin hold radius were 3 ft. the rotational speed of the centrifuge would be at least 312 r.p.m., corresponding to a periodic time of .19 seconds. If a continuous centrifuging system were designed in which skin entry could only take place within a limited range of angular positions of the rotating structure (Figure 3, p. 21), this range must correspond to a time period of less than .19 seconds. The accuracy of skin entry timing and high skin entry velocity required, excluded such systems from further consideration. Two alternatives remained, either that skin entry should be coaxial with the rotation of the machine (Figure 4, p. 21), or if offset, the skin should have an unrestricted annulus in which to fall (Figure 5, p. 21).

The hold time of the skin within the machine could be determined by the speed of driven moving parts within the rotating structure, external timing systems causing movement of parts within the rotating structure, or the time interval between consecutive skins. Systems in which the speed of driven moving



parts (e.g. conveyor belts) within the machine control hold time have the advantage that hold time remains constant irrespective of rate of skin entry, and that no external control systems or rotating couplings between centrifuge and control unit are necessary. Systems in which the release of a skin in the machine is caused by the ingress of a subsequent skin have the advantage of simple control over hold time, but require a constant skin feed rate for maximum throughput.

Removal of a sheepskin from a centrifuge could be achieved by removing restraints on radial motion of the skin and thus allowing tangential escape from the rotating system. The skin could then be brought to rest by a cylindrical wall containing the machine and mechanically collected and placed on an exit conveyor belt.

II.2 METHODS OF OPERATION

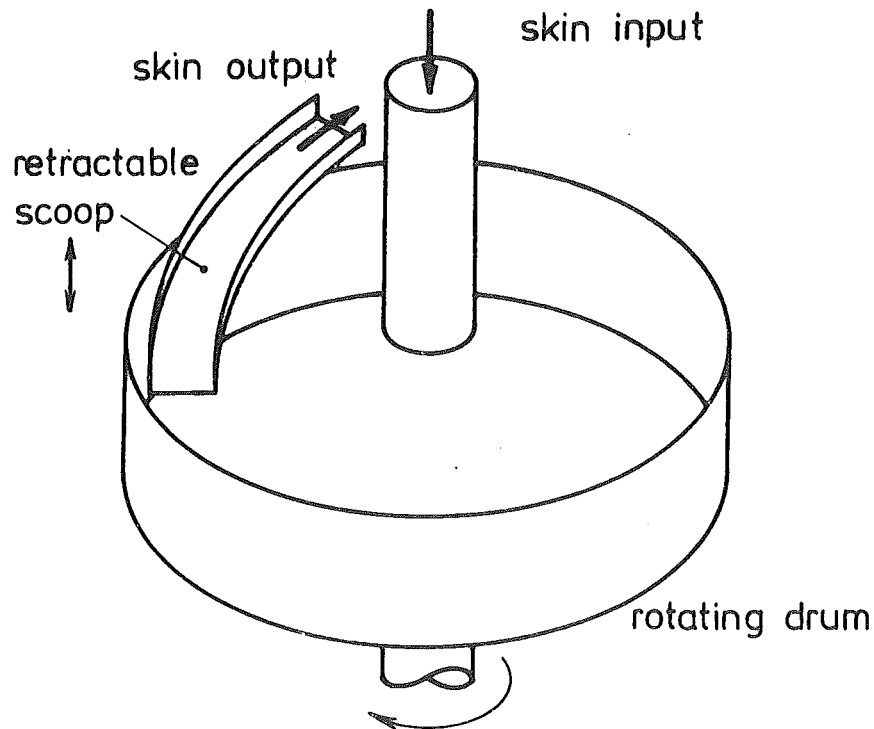
A previous investigation⁽¹⁾ into the possibility of a continuous centrifuge for woolly sheepskins resulted in the proposal of a machine using conveyor belts to pass the skins in an axial direction through a rotating drum (Figure 7, p.25). The difficulties involved in operating conveyor belts in high acceleration fields and the problems expected at the edges of the belts encouraged investigation into alternative systems.

Many different mechanical systems for achieving the required controlled hold time in a centrifugal acceleration field were considered. The following pages show diagrams of some of these systems with comments on the characteristics expected. When considering the operation of these systems it should be realised that a 50 lb sheepskin in a 100g acceleration field exerts in excess of 2 tons force on the circum-

ferential restraint.

Drive mechanisms for gates, scoops, belts, etc. are not shown as many configurations are possible for each system. Drain holes for escape of water from the rotating systems are not shown, but would be present in machine parts radially outwards from the skin hold positions, with suitable guides, if necessary, to ensure that water escape and skin escape occur in different planes.

FIG. 6. RETRACTABLE SCOOP

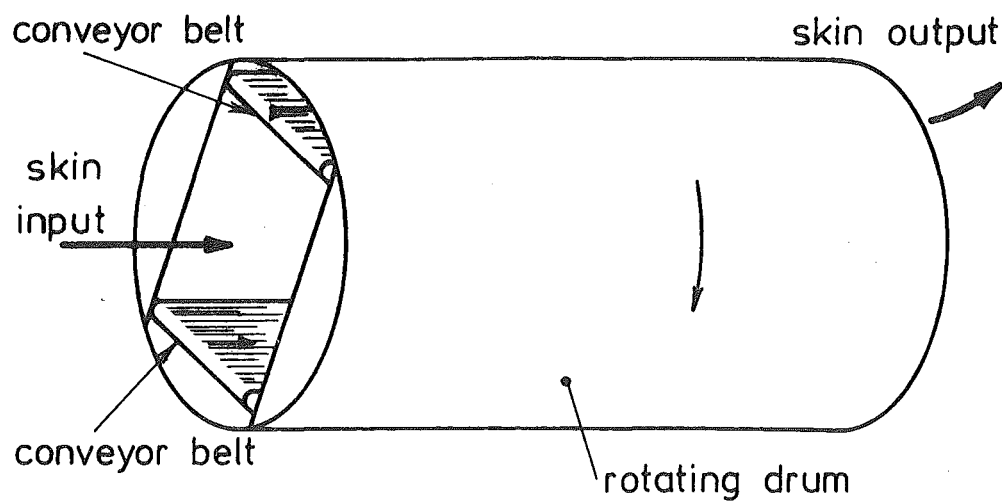


Operation: A skin fed in moves to the circumference. The hold time elapses. The scoop is lowered immediately after the skin has passed below it, and is in position with its wall and floor immediately adjacent to drum wall and floor in less time than is taken for one revolution of the drum. The skin is then discharged, scoop raised and cycle repeated.

Comment: Damage to skin probable (jamming between scoop and drum). Skin position sensing necessary.

Mechanical design problems : Small scoop to drum clearance necessary.
 High loading on scoop from skin.
 High loading on skin from scoop.
 High acceleration required of scoop.

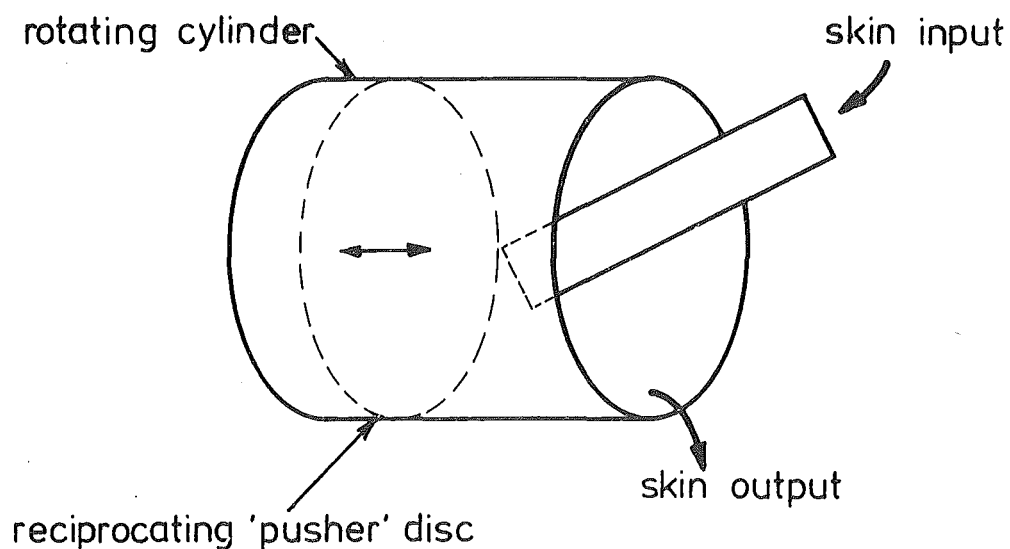
FIG. 7. CONVEYOR BELT. (PRINCIPLE SUGGESTED BY RANDS.)



Operation: Skins fed in are carried through the drum by conveyor belts driven in the direction shown, and discharged. The belt speed determines hold time.

Comment: Skin may enter gaps at edges of belts causing jamming and skin damage.

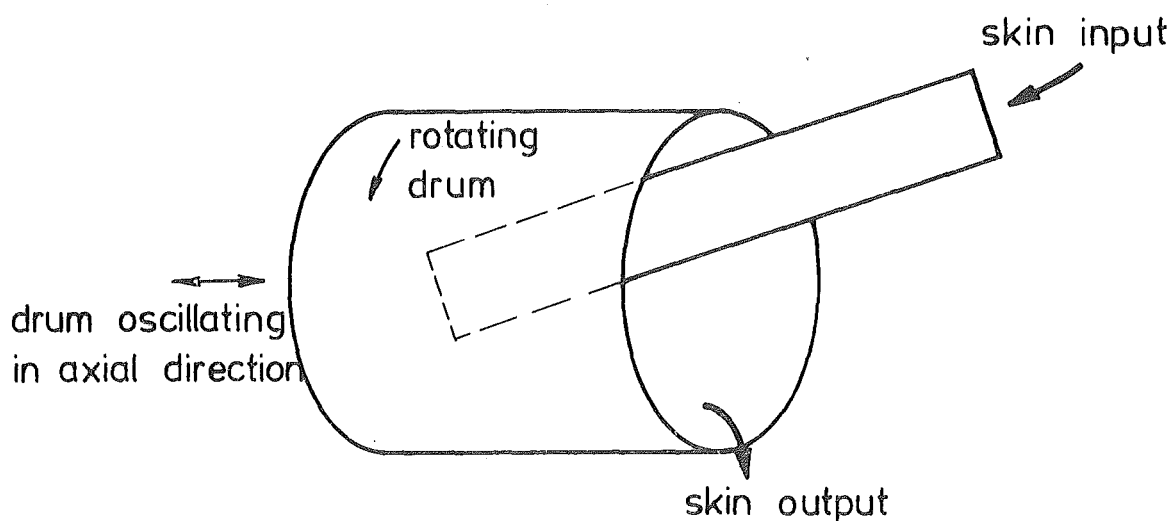
FIG. 8. PUSHER TYPE (SYSTEM USED SUCCESSFULLY FOR SOME PARTICULATE MATERIALS).



Operation: Skins are fed in adjacent to the pusher disc. Axial reciprocating motion of the disc causes a flow of skins towards the open end from whence they are discharged.

Comment: Skin damage and jamming probable.

FIG. 9. VIBRATORY FEED



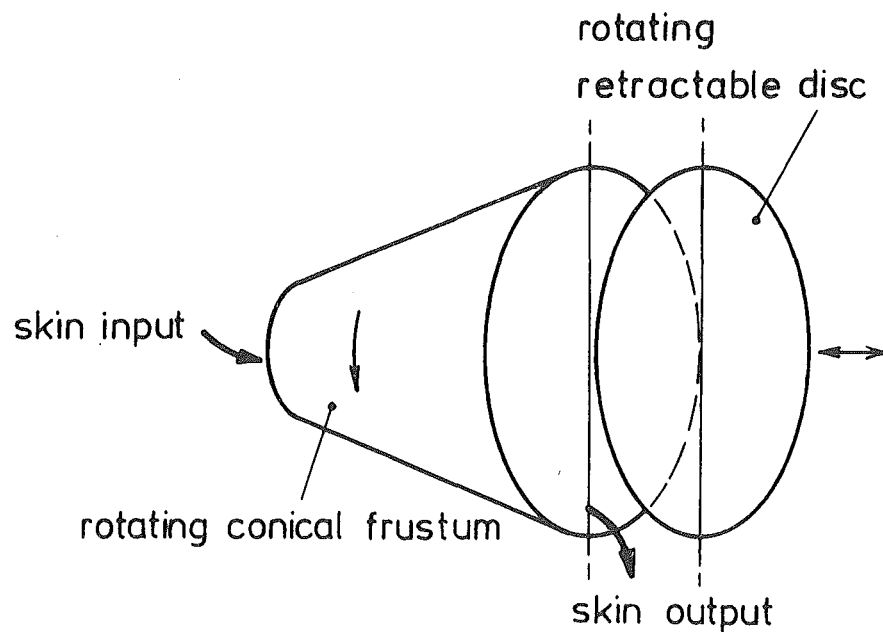
Operation: Oscillation of drum is asymmetric, peak instantaneous acceleration when the drum is moving to the left in the above figure exceeding that to the right. Skins fed into left hand end flow towards the right hand end in a series of small movements and discharge.

Comment: Peak instantaneous axial acceleration of the drum must exceed (drum material μ sheepskin) \times centrifugal acceleration field magnitude (more than 800 ft/sec^2).

Mechanically complicated.

Noisy operation.

FIG. 10. DISC GATED CONE.

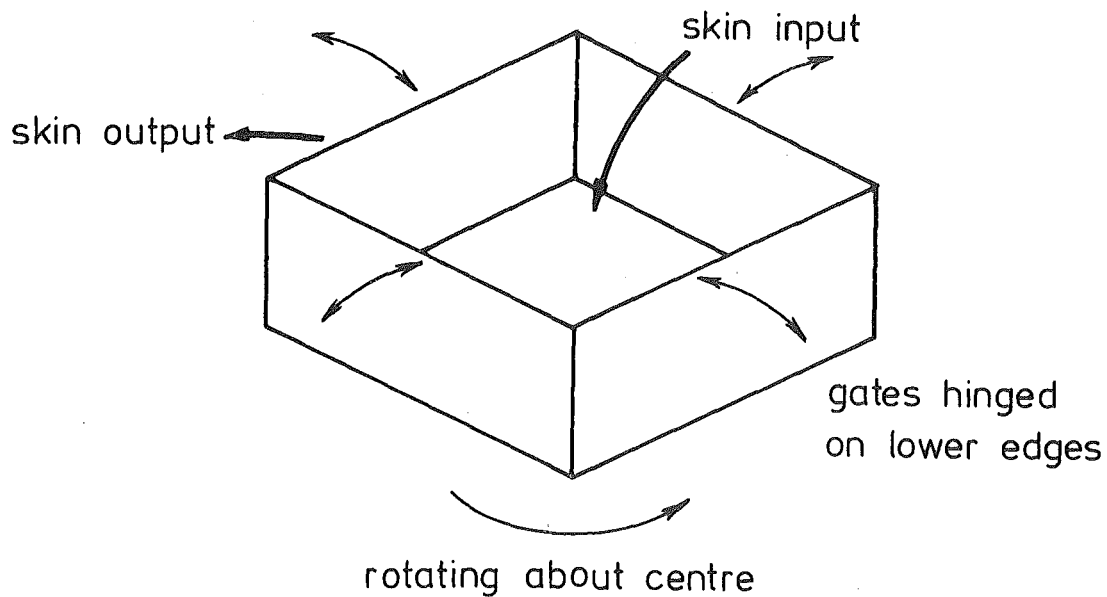


Operation: Skins fed in slide to larger diameter end of frustum. The disc is retracted after elapse of hold time, and the skins escape. The disc is replaced against the end of the frustum and the cycle recommences.

Comment: Separate mechanical support is required for the disc and the frustum.

Timing systems required.

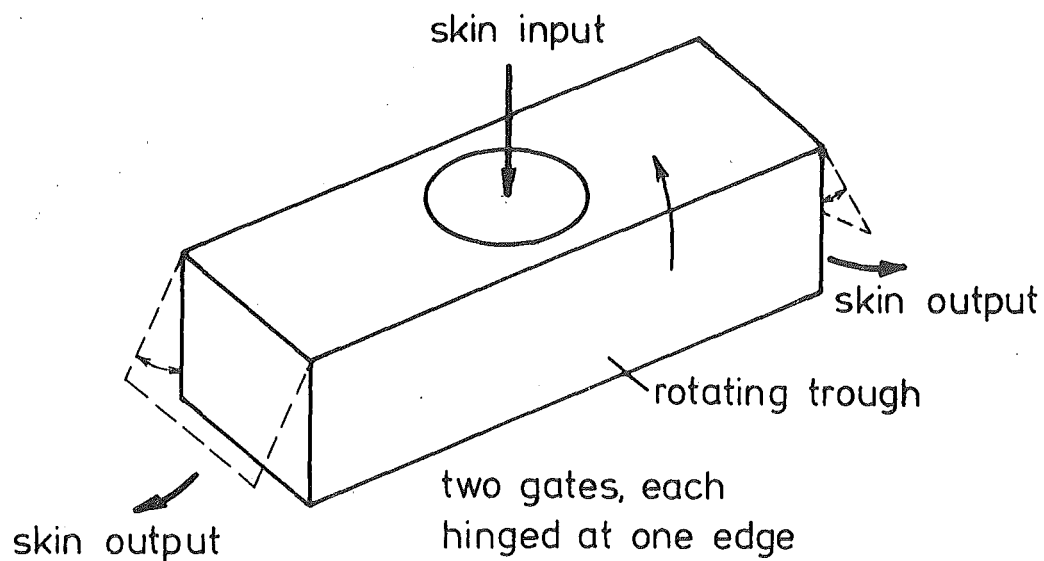
FIG. 11 CIRCUMFERENTIAL GATES.



Operation: Skins fed into the centre move to the periphery and are held by the vertical gates. After the required hold time has elapsed, the control system allows the gates to open and release the skins. The gates are closed and the cycle recommences.

Comment: Control system required for timing gate movement.
 Mechanically cumbersome - operation of gates
 - rotating hydraulic
 mechanical or electrical
 coupling.

FIG. 12. GATED TROUGH



Operation: Skins fed in centrally move to either end of the trough and are held by the gates. After elapse of hold time, gates are opened to allow skin escape and closed again. Cycle recommences.

Comment: Mechanically simpler than the circumferential gating system. External timing systems required.

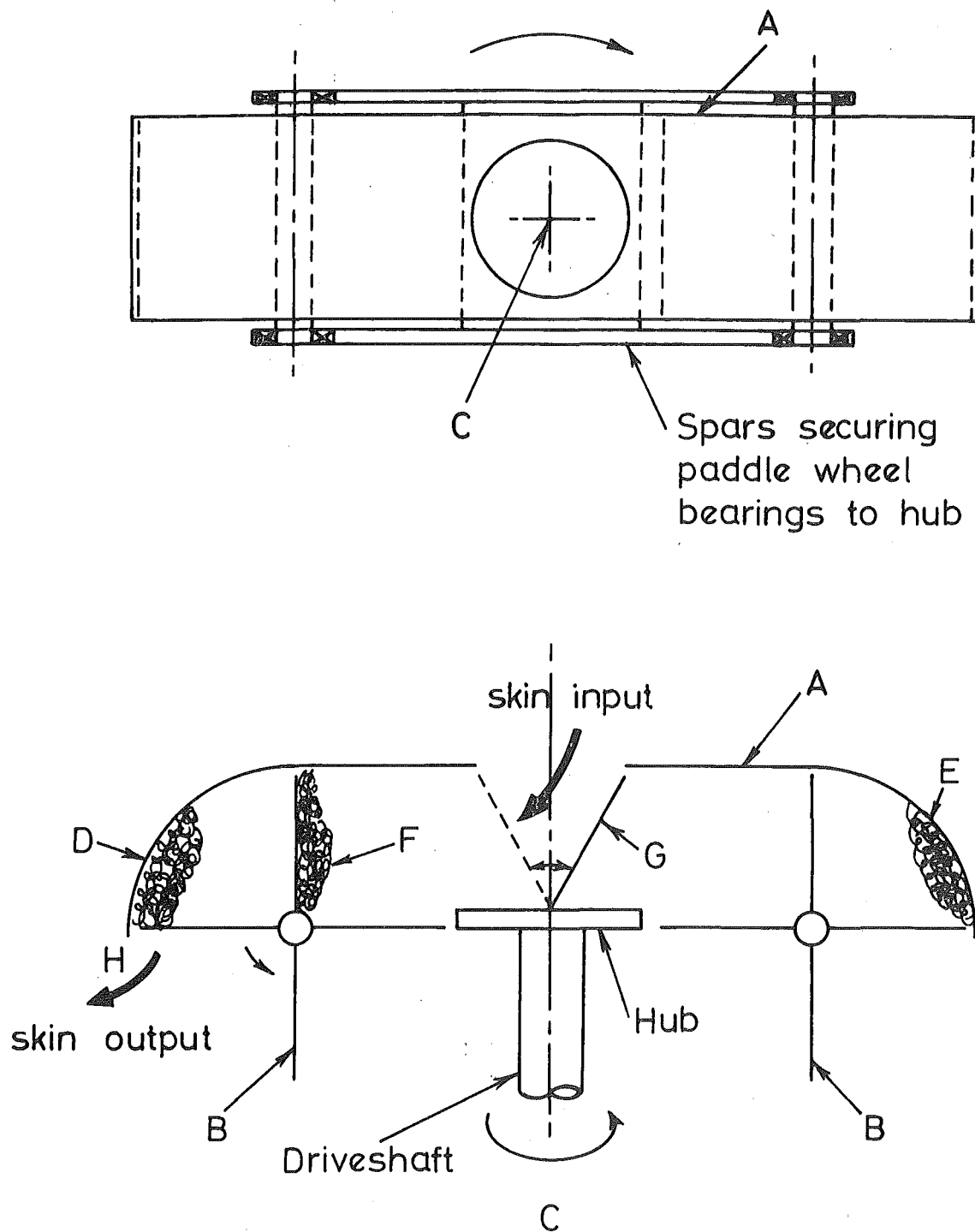


FIG 13 PADDLE GATED TROUGH
(PRINCIPLE SUGGESTED BY THE
AUTHOR IN UNDERGRADUTE
DESIGN STUDY)

Operation: (Refer Figure 13, p. 31). A trough-shaped casing (A) with a bladed wheel (B) at each end rotates about axis C. Each of these bladed wheels is free to rotate about its own axis. A central gate (G) pivoted along its lower edge is moved by linkages connected to both bladed wheels in such a manner that a 90° rotation of either wheel will move the gate between its extreme positions. Two sheepskins (D and E) are in the end compartments formed by the bladed wheels and the trough as a result of previous cycles of operation. A third skin, F, is introduced through the central hole in the top of the trough and directed by the gate (G). As a result of the acceleration field caused by rotation about axis C the skin (F) strikes the vertical paddle blade in the position shown. Consequent rotation of the paddle wheel allows the discharge of skin D and results in the movement of the central gate to the position shown by the dashed line. The paddle wheel continues to rotate until skin F occupies the position previously occupied by skin D. Rotational kinetic energy is dissipated during the latter part of this 90° rotation of the paddle wheel by friction between the skin and the curved trough end. If necessary, a cam indexing system could be devised to provide preferential angular positions for the paddle wheels, or interlocks provided to limit paddle wheel rotation to 90° for each cycle of the central gate.

Comment: No external timing or synchronising systems required other than those necessary to ensure time between alternate skin inputs is greater than the minimum hold time required.

II.3 CHOICE OF SYSTEM

The factors influencing the choice of system were divided into four categories:

1. Known or suspected problems involved in the handling system.
2. Mechanical complexity of the system.
3. Ease of alteration of hold time and acceleration field.
4. Throughput for a given hold time.

Anticipating less obvious problems with any one handling system mechanism could only be conjecture. While forces required for operating mechanisms and loading on machine elements could generally be calculated, the actual performance of many of these systems could only be determined by experimenting with hardware.

It was decided that a machine should be designed on the paddle-gated trough system.

II.4 MODEL CENTRIFUGES

Construction of a small scale model of the paddle-gated centrifuge was considered. To be of value, a model should provide information on the feasibility of the handling mechanism, and/or further information on suitable dwell time and acceleration field combinations for centrifuging sheepskins.

Rate of escape of water from a woolly skin is affected by restrictions on the passage of the water in the vicinity of the wool. Rate of water loss from the wool on a small piece of skin in a model centrifuge is unlikely to be the same as that for a complete skin in a full size machine.

Effective modelling requires that the effects of scale modifications are predictable. When a sheepskin is dropped into a rotating machine its behaviour is affected by at least two non-linear phenomena:

1. The modulus of elasticity of the skin varies with strain;
2. The dynamic coefficient of friction between a wet woolly skin and a steel surface varies unpredictably according to factors influencing the extent of fluid film separation of the materials.

The development of a model skin from which useful results could be obtained from a model centrifuge would in itself be a complex problem, outside the scope of this work.

It was concluded that useful information on water removal or handling mechanisms could not be obtained from a small scale model centrifuge.

CHAPTER THREE

DESIGN OF THE PADDLE-GATED CENTRIFUGE

The general form of the first experimental paddle-gated centrifuge is shown in Figure 14, p. 36.

For experimental purposes a feed conveyor and exit conveyor were unnecessary. Instead, a rectangular trough was fitted along which the skins could be pushed to fall into the entry guide, and an annular tray was provided onto which the skins would fall after release from the rotating parts of the machine.

Indexing and interlock systems for the paddle wheels were not initially fitted, as a better understanding of the requirements of these mechanisms was expected after experimentation with free running paddle wheels.

For similar reasons, central gating was initially omitted. Instead, curved guide plates (Drg.CC18) were fitted to the inside surface of the trough walls adjacent to the entry hole. The shape of these guide plates was such that a particle entering the trough and exhibiting the same friction characteristics as a sheepskin would slide to either end, but not remain against the side wall of the trough.

Guttering to separate the planes of discharge of water and sheepskin from the rotating structure was not initially fitted.

III.1 Machine Dimensions

Machine dimensions were primarily dependent on the maximum size of sheepskin to be processed.

Consultation with persons associated with fellmongery showed that design for a maximum wet skin weight of 50 lbs should be satisfactory.

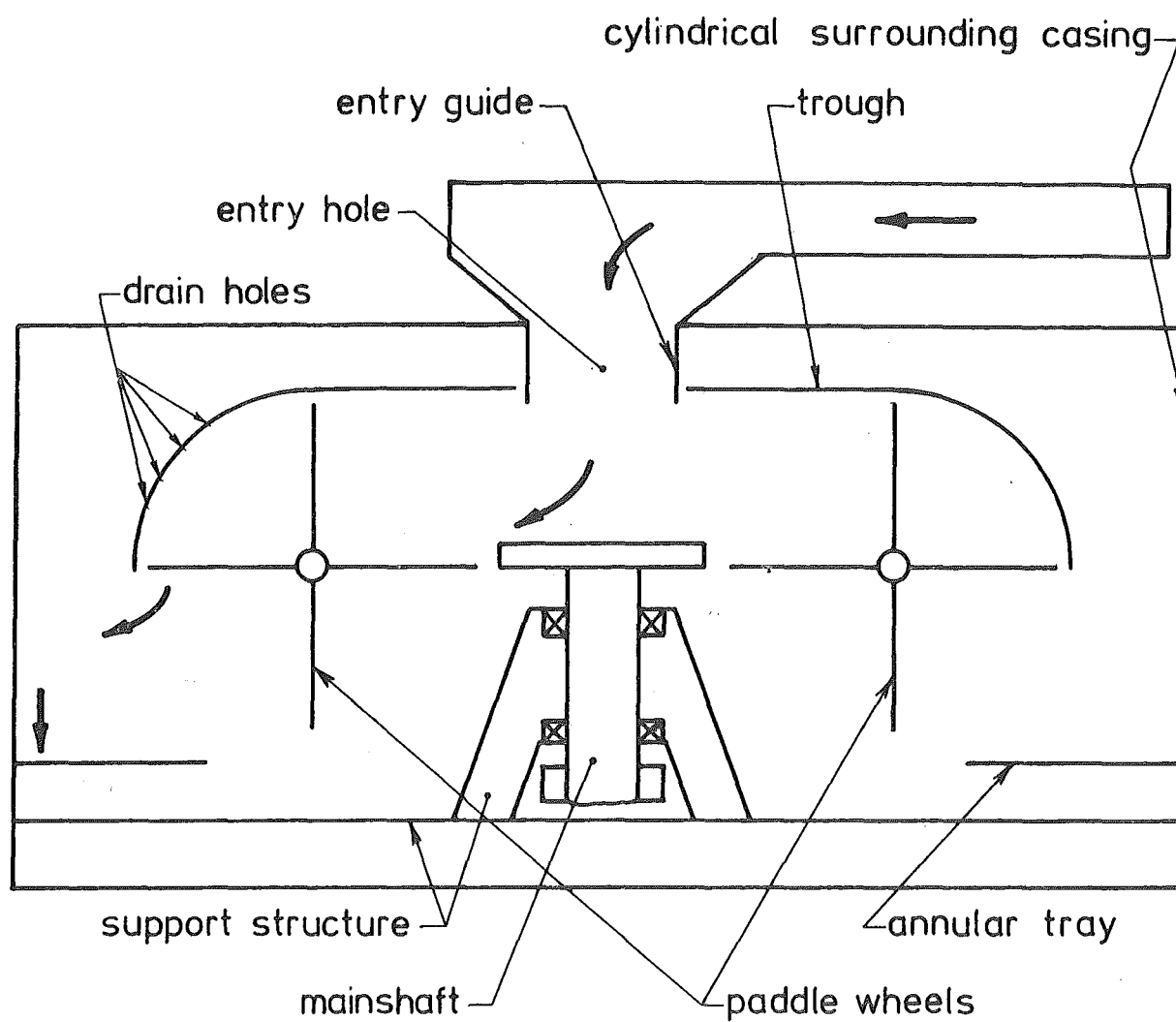


FIG. 14. GENERAL FORM OF THE FIRST PADDLE GATED MACHINE (SCHEMATIC CROSS SECTION).

Wet woolly sheepskins were weighed and measured and found to occupy approximately 35 cubic inches per pound. A maximum wet skin volume of one cubic foot was therefore assumed.

Tests were performed to determine the minimum size circular hole through which a wet woolly skin would pass under the influence of gravity. The largest skin available at the time had a wet weight of 35 lbs. This was found to pass easily through a 12 inch diameter hole under its own weight if a suitable conical frustum entry was provided. Experiments with various cone angles showed that this had little effect on ease of skin passage within the semi angle range of 15° to 45° . With semi-angles greater than 45° the skin became progressively more difficult to pass through the 12 inch diameter hole.

Centrifugal acceleration magnitude (α) at the skin hold radius depends both on angular velocity (ω) and machine radius (R), according to the relation $\alpha = \omega^2 R$. Identical centrifugal acceleration magnitude may be obtained either by rotating a small machine at high speed or a larger radius machine at lower speed. The choice of machine radius and rotational speed for one particular acceleration field magnitude affects both the tangential velocity at which the skin leaves the machine and the power required to overcome aerodynamic drag.

If R = centrifuge radius

k_0 is a constant

and V = centrifuge peripheral velocity,

then for one particular acceleration field magnitude

$$V = k_0 (R)^{\frac{1}{2}}.$$

Skin exit velocity is therefore lower for smaller machines. After leaving the rotating parts of the machine the skin strikes the cylindrical surrounding casing. It was thought that skin

damage could occur during impact, the extent of damage increasing with impact velocity. It was therefore considered that the machine should be as small as was practicable to reduce the possibility of skin damage.

The aerodynamic drag on a trough varies with radius in approximately the same way as drag on a rotating plate (see Figure 15, p. 39).

If δr is the width of a narrow strip height h at radius r , v is the tangential velocity of this strip, ω is the angular velocity and k_1 , k_2 and k_3 are constants, then for a particular acceleration field magnitude at R :

$$v = k_0 (R)^{\frac{1}{2}}$$

$$\text{and } v = \frac{R\omega}{r}$$

$$\text{i.e. } v = \frac{r}{(R)^{\frac{1}{2}}} k_0 .$$

$$\text{Aerodynamic drag for an element } \delta r \doteq h \cdot \delta r \cdot C_D \cdot v^2 \cdot k_1$$

$$\begin{aligned} \therefore \text{ Total aerodynamic drag power } &\doteq \int_0^R k_2 \cdot \frac{r^3}{(R)^{\frac{3}{2}}} \cdot dr \\ &= k_3 (R)^{2.5} \end{aligned}$$

i.e. aerodynamic drag power increases with increasing machine radius for any one particular centrifugal acceleration field magnitude.

The following machine dimensions were the smallest considered suitable for all skins and were used in the design of the paddle-gated trough machine :

Entry hole diameter	14"
Trough width	18"
Trough height	15"
Trough tip radius	36 $\frac{1}{4}$ "

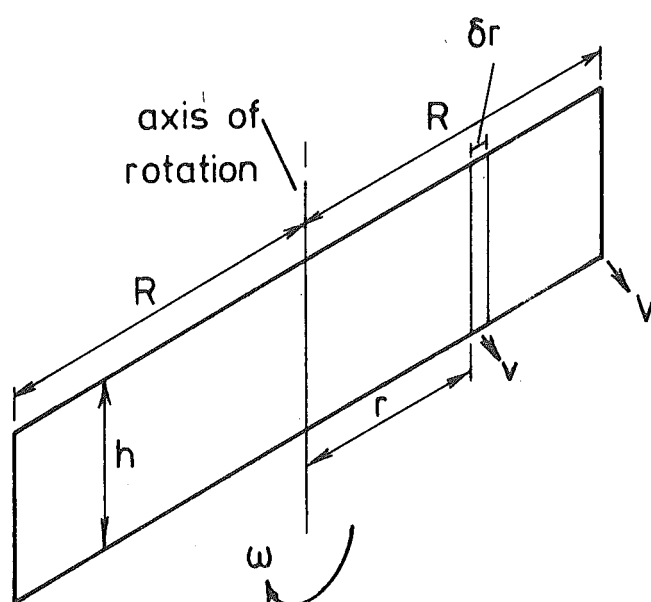


FIG. 15. AERODYNAMIC DRAG ON A ROTATING PLATE.

III.2 ROTATIONAL SPEED

Rands found from his experiments that centrifugal acceleration fields of approximately 140g could reduce wool moisture to the level obtained by conventional hydro extraction. Rands suggested that a hold time of five seconds or more should be sufficient at this acceleration.

Whereas it was expected that the geometry of a centrifuging system would affect water removal rate at any one acceleration field magnitude, machine geometry was not expected to significantly influence the level of wool moisture approached with increasing time for a given acceleration. Power consumption of a continuous centrifuge increases with increasing acceleration field magnitude. For identical mass throughput and machine dimensions the relationship between acceleration field (α) and power consumption (W) is approximately*:

$$W = \underbrace{A (\alpha)^{1.5}}_{\text{power to overcome aerodynamic drag}} + \underbrace{B (\alpha)}_{\text{power to accelerate skins to machine speed}}$$

where A and B are constants.

For similar machine dimensions the skin exit velocity, V, (and possibility of skin damage, see Section III.1) increases with increasing acceleration field :

$$V = k_5 (\alpha)^{\frac{1}{2}}$$

where k_5 is a constant.

Design for minimum effective centrifugal acceleration magnitude was therefore desirable for these as well as structural reasons.

Construction of experimental hardware for obtaining further information on the effect of varying hold time at

* because α is proportional to ω^2

various acceleration field magnitudes was considered.

Results from such an experimental system may not have been similar to those from a paddle-gated centrifuge unless both systems were identical in the following respects :

1. Capability to handle a full size skin.
2. Time for a skin to come up to machine speed after entry into the rotating system.
3. Restraints on the escape of water, including drain hole configuration and skin orientation during hold.
4. Skin deceleration time on leaving the rotating system.
5. Acceleration field magnitude.

An experimental system which would give useful results would have been of similar complexity and cost to a paddle-gated trough centrifuge.

It was therefore decided to design the paddle-gated machine for an acceleration field of 140g at the end of the trough, and to design the drive so that alteration of the speed of the centrifuge was possible.

III.3 STRUCTURAL MATERIAL AND STRESS

Low carbon steel construction was used both for economic reasons and for ease of welding, machining and forming. Steel conforming to BS 15 was readily available on the local market. This material has a minimum yield stress of 16 t.s.i. and conforms closely in composition to EN 3. The fatigue limit for EN 3 in water is given in "Properties of the EN Steels"⁽³⁾. A Soderberg diagram for $f_a = 8$ t.s.i. and $f_m = 16$ t.s.i. was used to determine suitable stresses for structural components made from this material.

The effects of stress raising geometries such as bolt holes and changes in section were allowed for by using published data⁽⁶⁾.

Stress calculations can be found in Appendix A. Where loading or structure stress/load relationship is determined by unusual methods (e.g. paddle blades) a more detailed explanation of the techniques used is shown in the section describing the design of that component.

III.4 PADDLE WHEELS

Curved paddle blades were used (Drg. CC17) as this geometry had advantages over straight parallel sided or tapering radial blades. The loading on a blade during skin impact is less for a curved blade, around which the skin slides, than for a flat blade. The use of a single curved plate to form the boundaries of one quadrant of the paddle wheel eliminated welded connections across the width of the blades at the roots.

Neither loading on the paddle wheels from skin impact, nor blade stresses for a given loading could be accurately predicted analytically. To determine suitable plate thickness for the paddle blades an approximate figure for impact load was calculated and a model paddle blade coated with stain sensitive lacquer loaded in a testing machine.

III.4.1 Loading

When a skin strikes a vertical paddle blade, the blade is loaded as a cantilever because of restraint on the rotation of the paddle wheel due both to its own inertia and also any control systems that may operate on the paddle wheel shaft. For structural design purposes it was necessary to estimate

the maximum load exerted on the paddle blade during this skin impact.

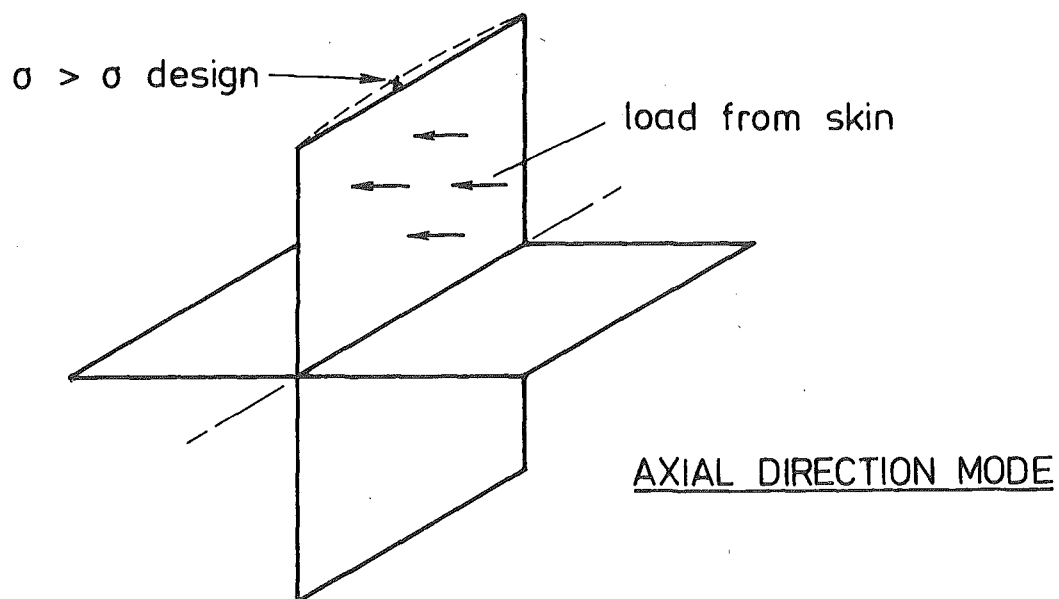
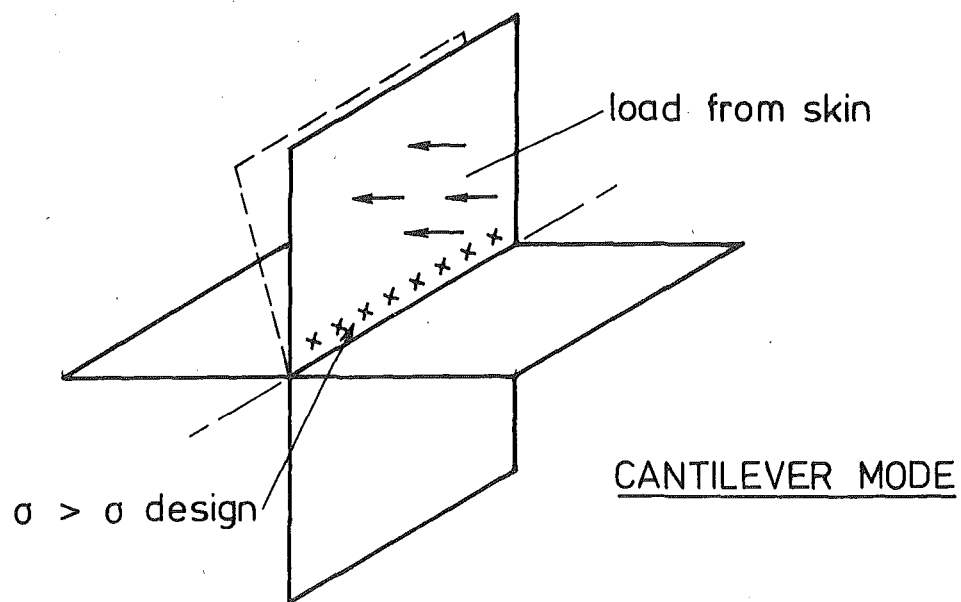
The velocity with which a skin strikes the vertical paddle blade was calculated, neglecting energy loss resulting from friction, to be 65 f.p.s. Making some allowance for friction, the actual impact velocity was estimated to be 50 f.p.s. Experiments indicated that the centre of gravity of the skin moves approximately four inches during impact. An impact load of 5,800 lbs was accordingly calculated for a 50 lb skin. The centrifugal acceleration force exerted by a 50 lb skin at the radius of the paddle wheel axis is 3,850 lbs. The peak loading would be less than the sum of impact load and apparent weight. A peak load of 7,000 lbs was estimated. (Measurements using a strain gauge telemetry system showed that the maximum load on the paddle blade was approximately ten percent greater than that due to the apparent weight of the skin, i.e. that the above estimate of loading was much higher than occurred in practice.)

III.4.2 Load-Strain Relationship of a Paddle Blade

Modes of failure considered for the paddle blades included failure as a cantilevered beam in the radial direction, and failure of the blade as a beam in the axial direction (see Figure 16, p. 44). The failure load as a cantilever was predicted using conventional beam theory. The load-strain relationship for the curved blade (for the axial beam mode of failure) could not be easily predicted analytically, so an experimental arrangement was devised for this purpose.

A full size single blade and web structure was fabricated

FIG. 16. TWO POSSIBLE MODES OF FAILURE OF THE PADDLE BLADE.



from 3/16" steel plate for use as a test specimen. Prior to loading, this test quadrant and six Hounsfield Tensometer test pieces were coated with "Stresscoat ST1205" brittle lacquer (see Appendix B) and dried at $75^{\circ}\text{F} \pm 5^{\circ}\text{F}$ for 24 hours. The test quadrant was then loaded in a Universal Testing Machine such that the curved plate experienced a similar distribution and direction of loading, and similar support, to that expected in the centrifuge.

At a load of 13.4 tons cracks appeared in the "Stresscoat" on the surface of the blade tip. Tests using the Hounsfield Tensometer and the lacquered test pieces showed that this corresponded to 1030 ± 50 micro strains. The relationship between load and maximum stress (assumed linear until yield occurred) was thus established for the test quadrant. Treating the paddle blades as thinwall structures (stress inversely proportional to plate thickness for a given loading) it was calculated that 1/8" plate would be sufficient to withstand the maximum calculated load of 7,000 lbs.

III.4.3 Preferential Positions of Bladed Wheels

It can be shown⁽⁷⁾ that the radial loading from a body on its supporting structure in a centrifugal field is equal to that which would result from an identical point mass at the centre of gravity of the body. Some bodies (e.g. a two-bladed rotor) when simply supported at their centre of gravity in a centrifugal acceleration field will tend to rotate towards a preferential angular orientation, i.e. will exert a torque about their centre of gravity except in particular equilibrium positions. Any such tendency by the paddle wheels would have affected machine operation.

It was shown that a wheel with four equally spaced identical blades has no such preferential positions. In Figure 17,

00' is the axis of rotation;

m is the mass of a small equivalent part of each of the four blades.

Then the total torque exerted about the bladed wheel axis by these four small masses is :

$$\frac{\delta m}{g} \omega^2 r \sum_{n=0}^{n=3} \left(\left[R + r \sin \left(\theta + \frac{n\pi}{4} \right) \right] \cos \left(\theta + \frac{n\pi}{4} \right) \right) \\ = 0 \text{ for all } \theta.$$

Any one blade is made up of a large number of such masses with a corresponding mass in each of the other three blades.

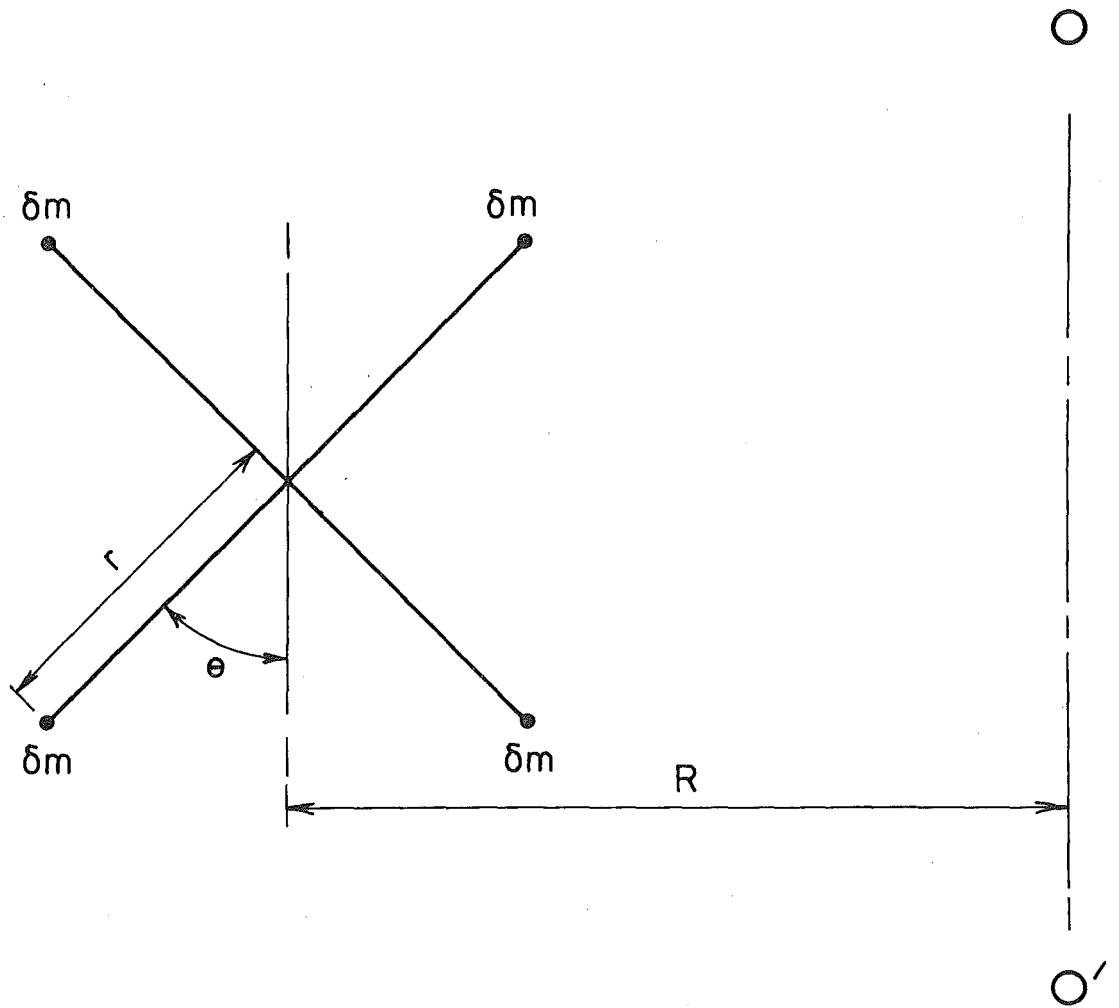
The four bladed wheel therefore has no preferential positions.

A similar proof shows that three bladed wheels (Section IV.2) also have no preferential positions.

III.5 TROUGH ENDS

After a skin strikes the vertical paddle blade and the wheel begins rotating, the centrifugal load of the skin is progressively transferred from the paddle blade to the curved end of the trough. When the skin is sliding on this curved surface, kinetic energy is converted to heat. Past a certain point during the rotation of the paddle wheel, the energy loss resulting from friction exceeds the energy input from the movement of the skin in the acceleration field, and the speed of rotation of the skin and paddle wheel relative to the trough decreases. At some subsequent point this rotation ceases, until a following skin strikes the next vertical paddle

FIG. 17. FOUR BLADED PADDLE STABILITY.



blade. The angle between paddle blade and trough at which this change from angular acceleration of the paddle wheel to angular deceleration (relative to the trough) occurs is a function of the coefficient of friction between the sheepskin and the curved trough end. For a point mass this change occurs when the angle between the acceleration vector and the normal to the trough end is $\tan^{-1}\mu$, (Figure 18, p. 49).

The coefficient of friction between both wet skin and wet wool and steel was determined by experiment. A wet woolly skin of known weight was dragged along a horizontal steel plate and the force required measured. It was found that greater force was required to drag a skin wool side down than skin side down, and that the coefficient of friction wool side down was in the range .5 to .8.

It was required that the curved end of the trough extend (as defined by angle θ Figure 18, p. 49) sufficiently to prevent the escape of the skin when the paddle wheel had stopped. Attempts were made to calculate a suitable value for θ for paddle wheels without indexing mechanisms. It was thought that practical experiments on the machine with different values of θ would give useful results in less time than would be required to develop a representative mathematical model.

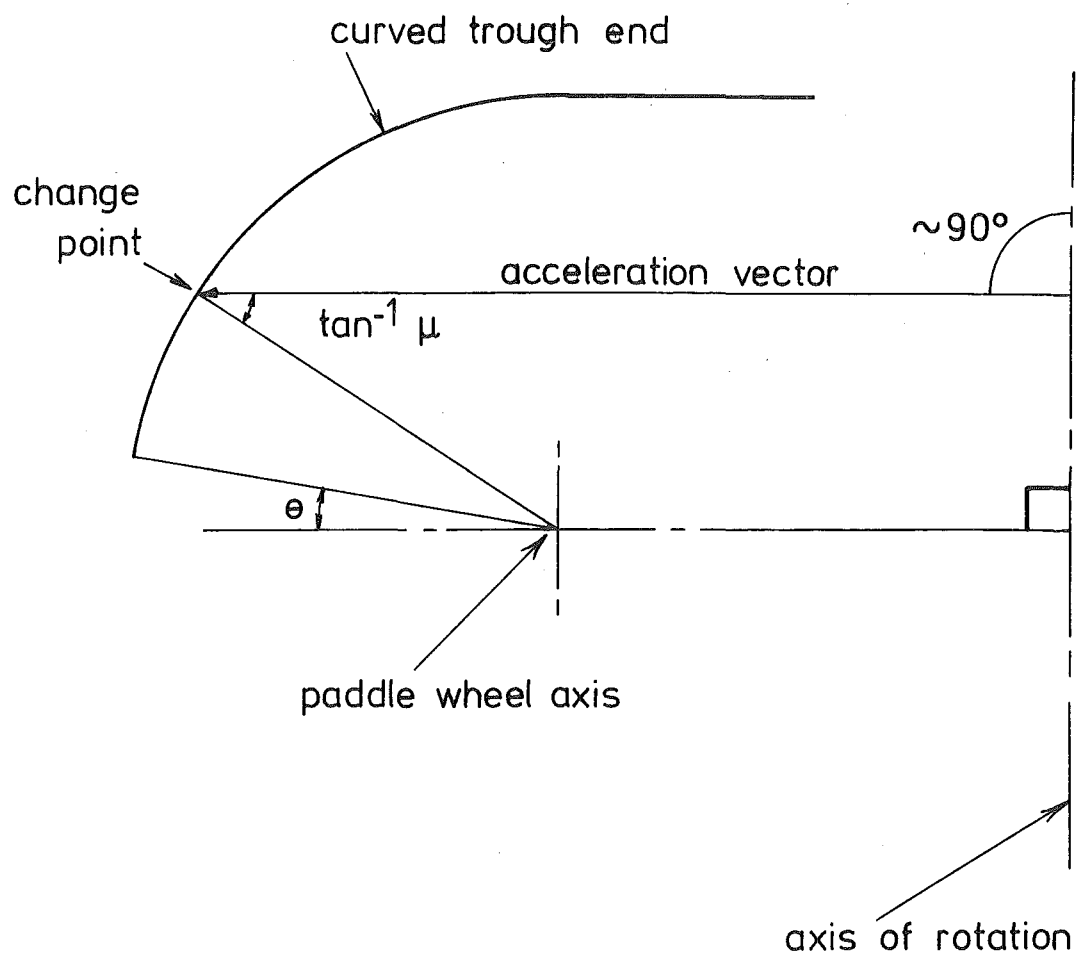
The machine was initially designed and built with $\theta = 20^\circ$.

III.6 DRIVE

The mechanical power required to drive the machine was calculated by adding expected aerodynamic drag, bearing friction, transmission loss and skin acceleration power consumptions.

Aerodynamic drag power requirement was predicted by two methods, one analytical, and one using scaling factors on

FIG. 18. CHANGE POINT



results obtained with a small model structure.

The figures obtained by these two methods were 17.5 hp and 18 hp respectively (see Appendix D). The good agreement between these estimates must be regarded as fortuitous in view of the assumptions which had to be made.

Rolling element bearings were used throughout the machine. Calculations using manufacturers' figures showed bearing loss to be negligible.

Transmission efficiency was estimated using manufacturers' data.

To accelerate 30 lb of sheepskin every five seconds to the machine circumferential velocity of 120 f.p.s. requires $2\frac{1}{2}$ hp. At this throughput $2\frac{1}{2}$ hp is required to move the skins in the radial direction, (average radial acceleration field of 70g, distance 3 ft).

Motor and Transmission (III.6.1): A 25 hp 3-phase electric motor was selected to drive the centrifuge via a single stage vee belt reduction. Predicted starting current with star-delta starting (120A) exceeded the capacity of the supply to the laboratory (60A). A "Twiflex" centrifugal clutch was used between the motor shaft and the driver sheave, which allowed the motor to run at 75% - 80% of synchronous speed during starting, and thus reduce starting current to within the supply capability.

For reasons given in Section III.2 the ratio of the vee belt drive was initially chosen to give a machine rotational speed of 380 r.p.m.

III.7 MECHANICAL SUPPORT OF THE MACHINE

Centrifugal drying machinery such as hydros and domestic

"spin dryers" are often designed to be "self balancing", in that the suspension of the rotating parts is of sufficiently low stiffness to allow these parts to rotate about their centre of mass at the operating speed. This method of mounting has an advantage over rigid mounting in that loading on the floor from out of balance forces is reduced. It was initially thought that this principle would be applied to the paddle-gated centrifuge.

The rotational speed at which self balancing occurs is necessarily above a certain transition speed at which forces undergo a 180° phase shift. When passing through this transition speed, large amplitude oscillations of the rotating system may occur. A rotating system which is axially symmetric within certain limits may be stable in a range of rotational speeds above this transition speed.

A rotating system which is severely axially asymmetric may have speeds or speed ranges above the transition speed at which the system is exponentially unstable⁽⁵⁾. The effect of periodic loads of unknown direction and magnitude (such as skins striking the paddle blades and trough walls) on such a system is not predictable. In view of the consequences of instability of the rotating system resulting in failure of the support structure, it was decided to arrange the support of the rotating system such that the machine operating speed was below the transition speed, i.e. "self balancing" was not attempted.

The bearings supporting the machine main shaft were secured to the floor of the laboratory via bearing housings, fabricated legs and the main base frame (Drg CC12 and CC13). These parts were provisionally designed for adequate strength (see Appendix A), and their combined stiffness calculated,

approximately, (see Appendix E) to ensure that the first critical speed of the system was above the proposed operating speed.

The critical speed of the machine was calculated to be more than eight times the operating speed.

CHAPTER FOUR

OPERATION OF THE PADDLE-GATED MACHINE

The machine was built in the form shown in Drg. CC26.

The machine was first run without sheepskins to check the performance of the drive, and to confirm that the machine did not pass through any critical speeds while accelerating to full speed.

Strain gauges were attached to the legs of the machine. Changes in leg strain were monitored by means of an oscilloscope connected to the output of the bridge. The peak value of the sinusoidally varying strain in the legs resulting from slight out of balance of the rotating structure (an inevitable consequence of manufacturing irregularities) was found to increase smoothly as the machine accelerated from rest to full speed. This confirmed that the machine was not passing through resonant speeds which could render invalid the stress analysis used in design.

Some problems initially experienced with "grabbing" in the centrifugal clutch were cured by reducing the coefficient of friction between the lining material and the drum with graphite powder, thus reducing the positive feedback effect of the leading shoes.

Both paddle wheels were locked with one blade vertical by wedges between the blades and the trough. Strain gauges were bonded to the spars (Item 53a, Drg CC16) on one end of the trough. These strain gauges were connected to a telemetry transmitter mounted on the rotating structure (see Appendix B). A small skin was dropped into the running machine several times and each time the output of the telemetry receiver (a strain analogue) observed on an oscilloscope.

When the skin went to the end of the trough which was strain gauged, the effect of skin impact on the strain in both the leading and trailing spars was found to be small. A typical graph of spar stress against time is shown in Figure 19, p. 55).

Test runs were then made with wet woolly sheepskins. Operation of the machine was not satisfactory. Sections IV.1, IV.2 and IV.3 describe the problems, and the modifications undertaken in attempts to remedy the problems.

IV.1 JAMMING OF THE PADDLE WHEELS

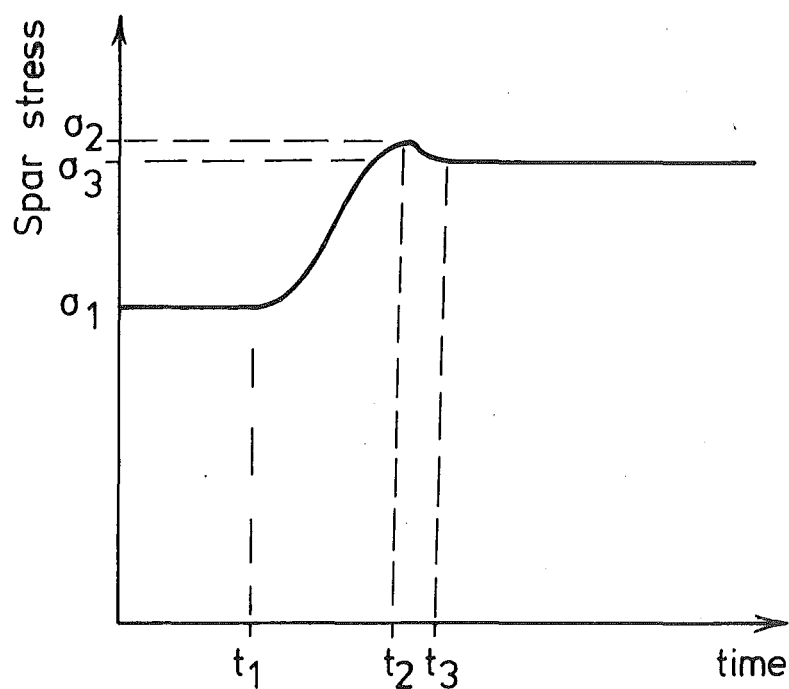
Small pieces of skin or wool sometimes became jammed between the paddle blades and the trough, preventing further rotation of the paddle wheel. On occasions partial disassembly of the trough and paddle wheel structure was necessary to free the paddle wheel. Parts of the woolly skin entered the small gaps between both the paddle tip and the curved trough end, and the paddle edge and the trough wall.

The clearance between the paddle blades and the trough was in the range $1/8"$ to $3/16"$ when the machine was first built. A second pair of paddle wheels was made and fitted as described in Section IV.2, with smaller clearances between the blades and the trough ($1/32"$ to $1/8"$). Jamming problems continued with the second set of paddle wheels with insignificant change in severity and frequency.

IV.2 PADDLE WHEEL CONTROL

In order that the rotation of the paddle wheel cease after $\frac{1}{2}$ revolution as was required for one machine cycle, two conditions must be satisfied :

FIG. 19. SPAR STRESS DURING SKIN IMPACT AGAINST A LOCKED PADDLE BLADE.



t_1 Leading part of skin strikes blade.

t_2 Peak loading on blade.

t_3 Skin retained by paddle blade.

σ_1 Spar stress resulting from centrifugal acceleration of machine parts.

σ_2 Peak spar stress.

σ_3 Spar stress resulting from centrifugal acceleration of machine parts and skin.

1. The kinetic energy of the paddle wheel and skin must be dissipated either by skin friction against the curved trough end, or by a combination of skin friction and a paddle wheel control system.
2. After the 90° rotation the total torque exerted on the paddle wheel from skin load on the blade and paddle wheel control systems must be zero.

As described in Section III.5 the angle θ (Figure 18, p. 49) influences the paddle wheel control systems necessary.

With $\theta = 20^\circ$ and without any paddle wheel control systems test runs showed that neither conditions 1 or 2 above were satisfied.

A spring loaded cam indexing mechanism (Drg CC22) was fitted to each paddle wheel shaft. A 5 lb skin placed in the end of the trough (position E, Figure 13, p. 31), when the machine was stationary, was ejected when the machine was brought up to operating speed. Condition 2 above was therefore not satisfied, i.e. the skin exerted more torque on the paddle wheel than that exerted by the spring loaded cam (approximately 200 lbs.ft). This was unexpected and could only be explained if the tests for coefficient of friction between wool and steel (Section III.5) gave misleading results. Further friction experiments were performed, this time loading the skin with weights to achieve interface pressures comparable to those existing in the centrifuge (5 to 50 p.s.i.) The coefficient of friction in this range was less than half that previously found at lower interface pressure, viz: .22 to .28 compared to .5 to .8.

The curved trough ends were extended to the plane of the paddle wheel axes ($\theta = 0$, Figure 18, p. 49). However, this had little effect on the performance of the system, skins still passing straight through the machine.

The curved trough ends were extended again, this time to 4" below the plane of the paddle wheel axes ($\theta = -15^\circ$, Figure 18, p. 49). In order that the following paddle blade was in the correct position to receive a new skin while its predecessor was being retained in the end of the trough, it was necessary to change the paddle wheel geometry. Two three-bladed paddle wheels were constructed and fitted (Drg CC24). Clearance between the new blades and the trough was smaller than that for the four bladed wheels. A new cam indexing mechanism (Drg CC25) was fitted as the original design was not suitable for three-stop indexing.

With this configuration machine operation was erratic. Some skins rotated the paddle wheel the required 120° , and others more or less. Only occasionally was a skin held, then discharged by the following skin.

IV.3 CENTRAL GATING

Occasionally a skin entering the machine would move completely towards neither end of the trough, but spread out along the length against one wall. If a second skin were fed into the machine on top of this first skin, and the second skin moved to one end of the trough, the consequent paddle wheel rotation usually dragged the first skin towards the same end. Because the orientation of the paddle wheel had changed from that required to receive a skin, different parts of the first skin often landed in different quadrants of the paddle wheel. The result was jamming of the paddle wheel.

Experiments were conducted on gating systems for positively directing the sheepskin to one end of the trough. Flat plates were placed in a number of positions (Figure 20, p. 59) in the trough adjacent to the entry hole and skins fed into the machine. Although all these configurations reliably directed a skin when the centrifuge was stationary, the dependence on skin geometry during entry of forces resulting from rotation of the machine made operation inconsistent.

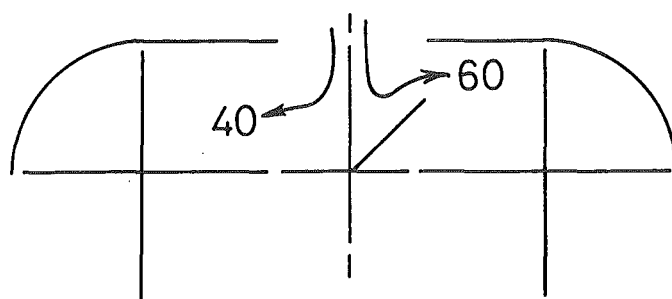
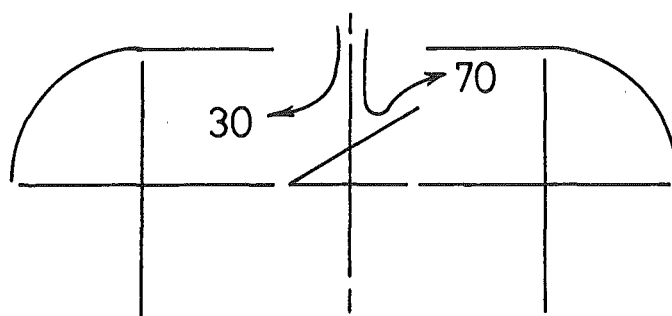
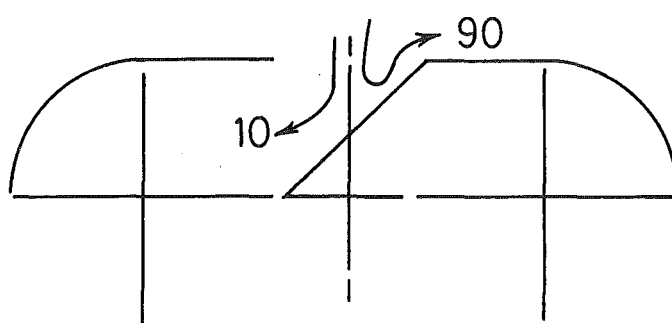
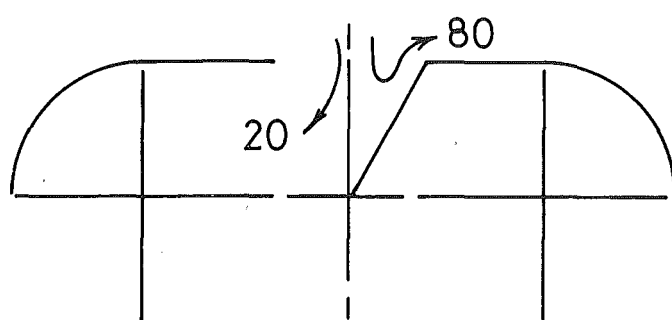
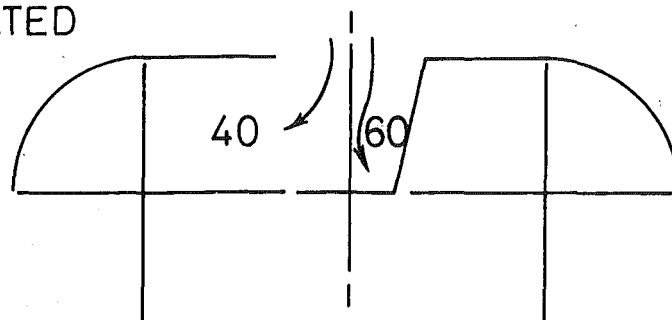
IV.4 DISCUSSION

No solution was known for the problem of jamming of the paddle wheels. A paddle wheel control system, including both indexing and energy dissipating, was apparently necessary. Directing of the skin to a particular end of the trough proved much more difficult than initially anticipated. It was decided that alternative systems should be investigated.

The performance of the paddle gated trough system contributed to the qualitative understanding of sheepskin behaviour under the influence of dynamic forces, which was of value in attempting to foresee problems which could occur with other centrifuge handling mechanisms.

Sliding seals were considered impractical. In the case of the paddle-gated trough, minimum clearance between the sliding parts was dictated by the irregularities in the trough inevitable in a fabricated structure. Wool entered gaps which at no point in the paddle wheel rotation were greater than $1/32$ ". There was, therefore, no reason to expect that reducing the clearance to this maximum between all mating seal parts in all paddle wheel positions by more accurate construct-

FIG. 20. CENTRAL GATING EXPERIMENTS. FIGURES INDICATE PERCENTAGE OF TOTAL OF TEN SKINS FOR EACH GEOMETRY TAKING THE PATH INDICATED



ion would eliminate jamming.

The concept of the skin as a coherent mass was not applicable in an environment subject to high acceleration. Some of the experimentally found results (in particular, the angle θ , Figure 18, required for Condition 2, Section IV.2, without paddle control systems) could only have been predicted if it had been assumed that the skin acted as a fluid.

Central gating was considered undesirable. While systems were possible for positively performing the central gating function by means of driven sliding structures, the associated sliding surfaces were considered undesirable because of the possibility of jamming by wool or skin.

CHAPTER FIVE

EXPERIMENTS WITH AXIALLY SYMMETRIC CONFIGURATIONS

The author was at this stage in a similar position to that prior to the conception of the paddle-gated trough system. Existing continuous centrifuging systems were unsuitable for sheepskins, and no mechanism could be imagined which was considered suitable for performing the mechanical handling required without external control and timing systems. It was thought that there may be a neater solution to the problem than a system such as the circumferentially gated machine shown in Figure 11 (p. 29) and its associated control systems.

The approach to the problem of conceiving a suitable handling mechanism had been modified as a result of the performance of the paddle-gated machine. Mechanisms employing sliding gates or seals were considered undesirable because of the possibility of jamming by wool or skin. Central gating had been found to require a more complicated mechanism than a simple inclined plate. The reduction in coefficient of friction between steel and sheepskin found with increasing interface pressure influenced the evaluation of systems for controlling skin motion.

An axially symmetric centrifuging machine with peripheral gating devoid of sliding seals would avoid the problems experienced with the paddle-gated machine. It was thought that practical experiments with such systems would give further insight into qualitative aspects of skin behaviour in high acceleration fields, and could prompt inventive imagination towards new ideas for a continuous centrifuging system.

For reasons explained in Section II.4 (p. 33) small scale model centrifuges are of limited value because the performance of other than full size machinery may not be representative of full size machine operation. The support framework and drive system were suitable for rotating structures other than the trough and paddle wheel assembly.

It was therefore decided to replace the trough and paddle wheel assembly with an axially symmetric structure capable of handling full size skins, both for observation of skins falling onto a rotating plate, and for observation of the effects of additional experimental hardware.

V.1 THE CONICAL FRUSTUM

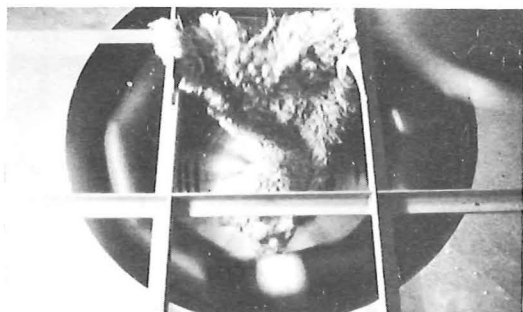
The trough and paddle wheel assembly was removed from the mainshaft hub and replaced with a conical frustum (Drg CC27). The cone semi-angle was greater than the angle of friction of a sheepskin on steel at the interface pressure expected. A skin fed into the rotating frustum was, therefore, not retained by the sloping wall.

A pneumatic tube was attached to the inner surface of the frustum (Figure 21, p. 64). When inflated, this tube prevented skin escape.

The tube was inflated and the motion of skins dropped into the centre of the rotating frustum photographed at 64 frames/second from above. A selection of frames from a typical run is shown in Plate 1. (see p. 63).

The motion of the skins was divisible into six stages :

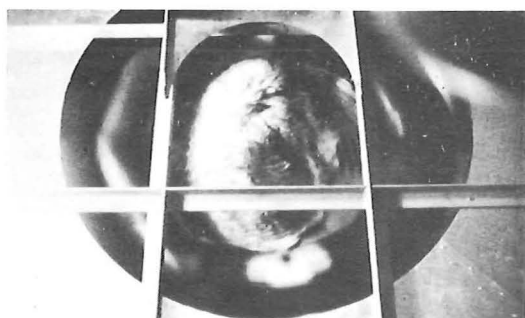
1. Initial contact of the leading part of the skin with the rotating base plate (Frame 1).
2. Rolling up of the skin about the centre of rotation of the machine (Frame 2).



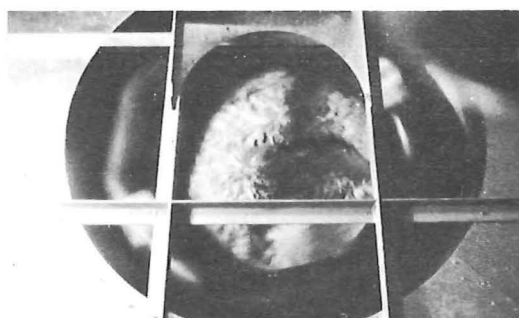
1
time = t_0



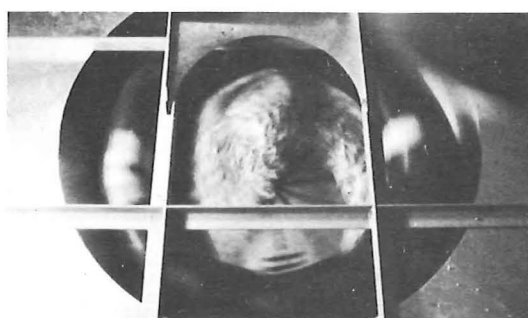
2
 $t_0 + 0.33$ sec.



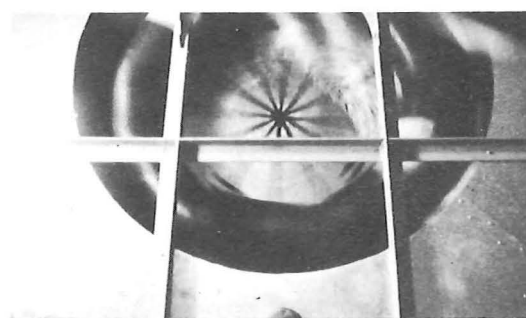
3
 $t_0 + 0.42$ sec.



4
 $t_0 + 0.58$ sec.



5
 $t_0 + 0.77$ sec.



6
 $t_0 + 1.36$ sec.

PLATE 1. SKIN ENTERING THE FRUSTUM

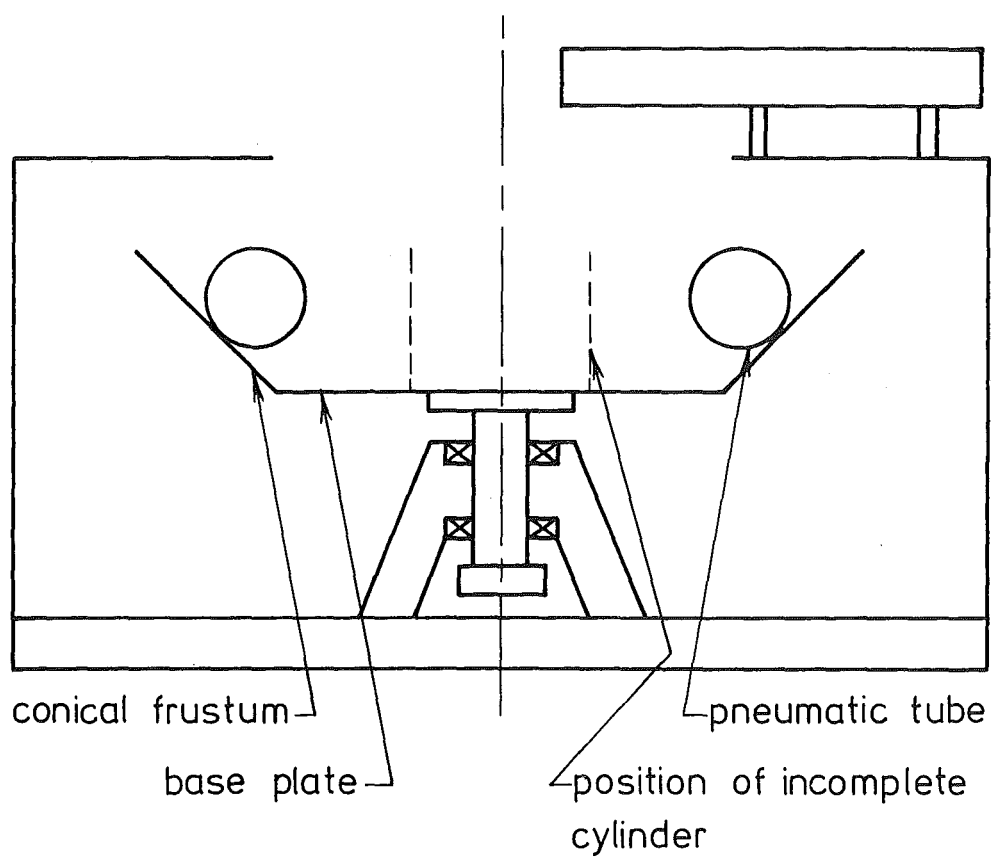


FIG. 21. GENERAL FORM OF THE FRUSTUM CONFIGURATION. (SCHEMATIC CROSS SECTION.)

3. Angular acceleration of the rolled skin as a coherent body (Frame 3).
4. Further angular acceleration of the skin accompanied by unrolling of the skin (Frame 4).
5. Radial movement of individual skin parts, stretching the skin in the circumferential direction (Frame 5).
6. Impact of the stretched-out skin on the frustum wall and pneumatic tube (Frame 6).

Several films were taken of skins entering the frustum. Skin entry and acceleration in the rotating system always followed the same general pattern, although the time between initial contact and adhesion of the skin to the frustum wall was reduced by non-coaxial introduction of the skin.

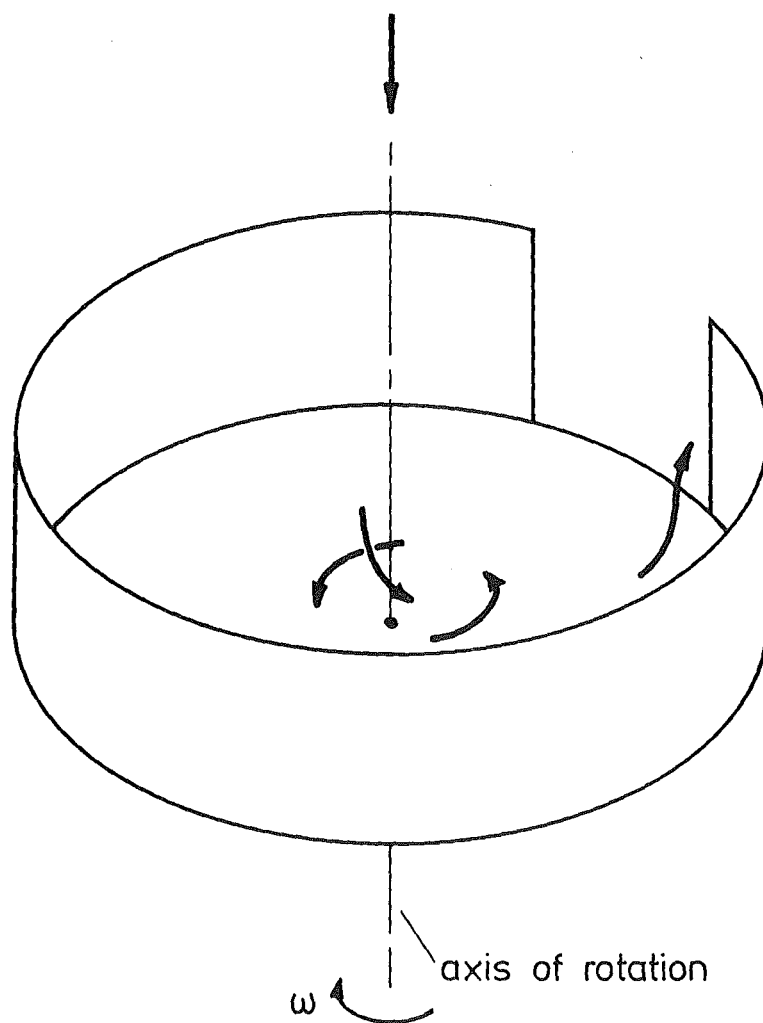
V.1.1 The Incomplete Cylinder

It was thought that a skin dropped into a rotating cylinder may slide around the cylinder wall more than one complete revolution before reaching machine rotational speed, and would discharge if a gap were left in the wall (Figure 22, p. 66). This would have been a useful mechanism for directing skins subsequent to entry into a rotating system.

An incomplete cylinder, 11" high, 18" diameter with 12" of its wall missing was mounted in the frustum machine coaxial with the frustum.

Skins were fed singly into the cylinder while the machine was running. Some skins discharged through the gap in the cylinder wall into the frustum as expected, but others remained inside the cylinder. Feeding further skins on top of those remaining in the cylinder sometimes resulted in discharge, but often skins accumulated to the extent that a further skin would not enter but be ejected over the top of the cylinder wall.

FIG. 22. INCOMPLETE CYLINDER.



The consequences of mounting the cylinder with the cylinder axis parallel to, but offset from, the centre of rotation were considered. This led to a new approach to the problem of devising a suitable mechanical handling system.

CHAPTER SIX

ECCENTRICALLY MOUNTED SYSTEMS

If a particle is dropped into a cylinder which is rotating about an axis offset to and parallel with the axis of the cylinder (Figure 23, p. 69), the positions at which the particle will come to rest relative to the cylinder may be limited to the centre of rotation and certain parts of the cylinder wall. If the rotational speed of the system is high (i.e. if gravitational acceleration is negligible), then a particle having a coefficient of friction μ against the drum wall will not remain against a point on the drum wall where $\tan \theta$ (Figure 24, p. 69) is greater than μ , but will slide around the drum wall towards region B.

It can be shown that θ has a maximum value (when $\phi = 90^\circ$) of :

$$\theta_{\max} = \sin^{-1} \left(\left(\frac{e}{r} \right) \sin \phi \right)$$

If $\tan \theta_{\max}$ is greater than μ , the particle will eventually come to rest relative to the cylinder either on the centre of rotation (unlikely in practice), or in region A or region B where $\tan \theta$ is less than μ .

It was thought that a skin dropped into such an eccentrically mounted drum would, if θ_{\max} were sufficiently high, slide to one of these two regions, A or B, and remain there.

The effect on the final position of a particle dropped into an eccentrically mounted rotating drum, of simultaneous slow rotation of the drum about its own axis, was considered (Figure 25, p. 70). It was concluded that the particle would move to and remain on a point on the moving drum wall,

FIG. 23. ECCENTRICALLY MOUNTED HOLLOW CYLINDER.

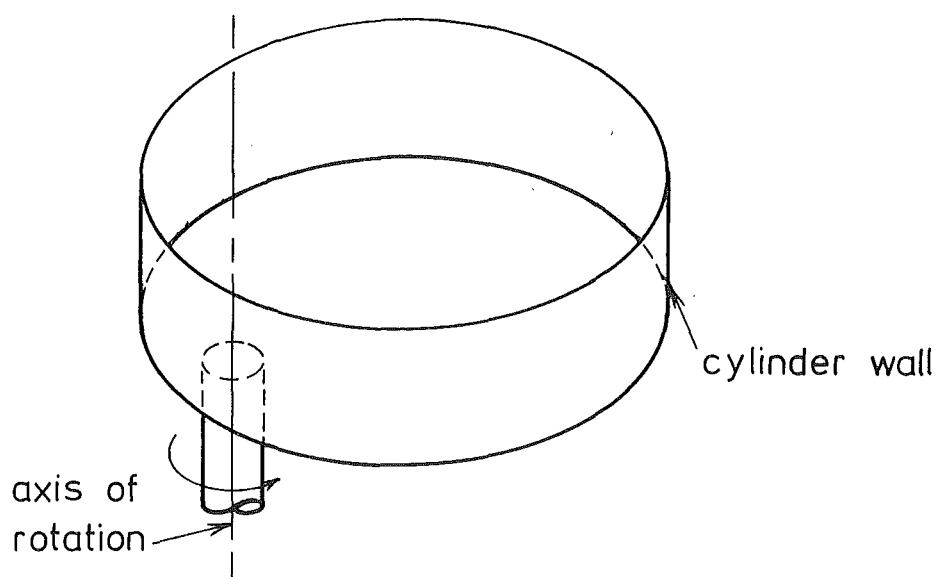
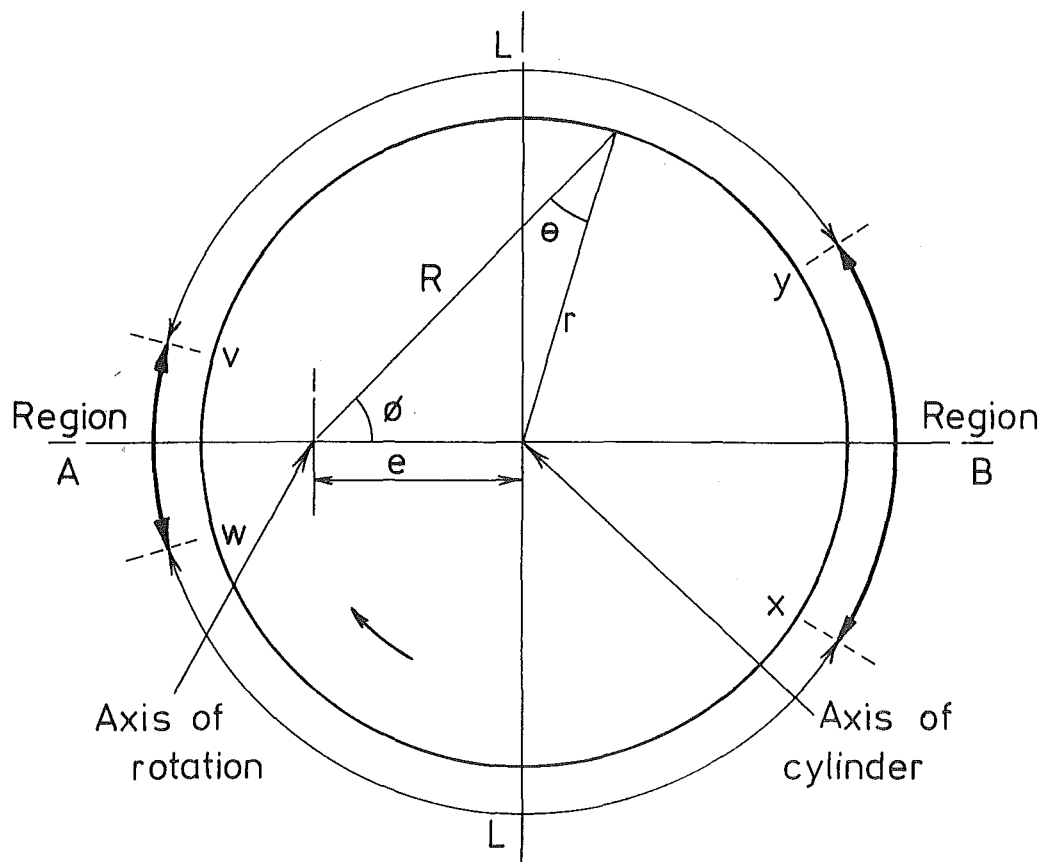


FIG. 24. PLAN VIEW OF ECCENTRICALLY MOUNTED CYLINDER.



$$\theta = \tan^{-1} \mu \text{ at points } v, w, x, y.$$

where $\theta = \tan \mu$, as shown in Figure 26. Although the drum wall would be moving relative to the particle, the particle would remain in this position relative to the line joining the two centres of rotation of the system.

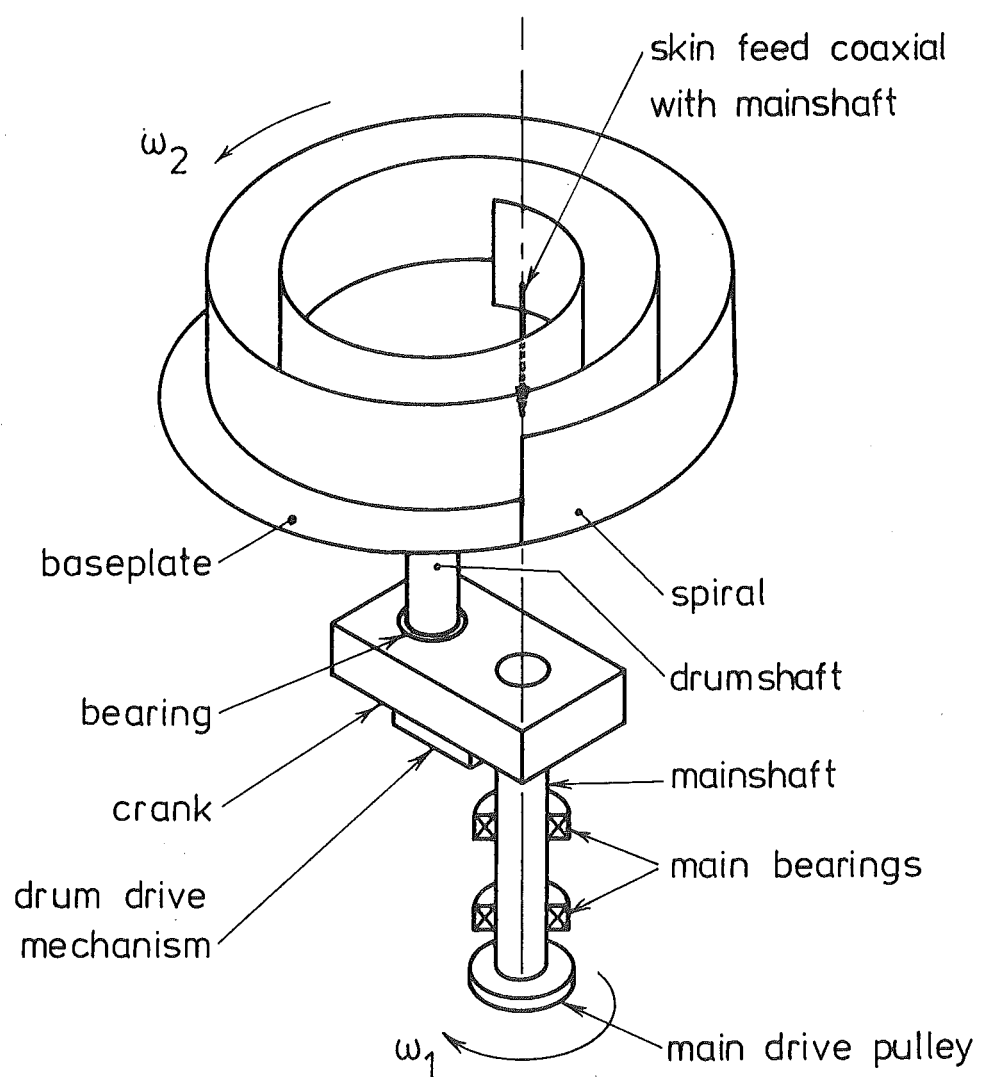
It was thought that replacement of the cylinder with an open ended spiral could result in a mechanical handling system capable of holding the particle for a time in the centrifugal acceleration field, and then discharge it. This "eccentric spiral system" would have none of the operating problems of the paddle-gated trough system, and would avoid problems associated with moving parts within the region of confinement of the material being centrifuged.

VI.1 THE ECCENTRIC SPIRAL SYSTEM

A machine operating on the eccentric spiral principle could take the form shown in Figure 27, p. 72 . The mainshaft and crank are rotated at a speed ω_1 . The spiral drum is rotated relative to the crank at some lesser speed ω_2 , by a drum drive mechanism. The eccentricity e (the distance between the mainshaft axis and the drum axis) is smaller than the inner radius of the spiral drum, so that the object to be centrifuged can be dropped into the rotating system coaxially with the mainshaft irrespective of the orientation of the drum at the time. The directions of rotation of the mainshaft and drum, and the "hand" of the spiral are arranged as shown, so that the initial motion of the object being centrifuged relative to the drum prevents the object entering the spiral until it reaches machine speed.

The acceleration field which the object is subject to during centrifuging depends on machine size and geometry, the

FIG. 27. AN ECCENTRIC SPIRAL SYSTEM.



mainshaft speed ω_1 and the coefficient of friction between the object and the spiral wall.

The time the object remains in the centrifuge after coming up to machine speed depends on the drum rotational speed relative to the crank, and the number of turns in the spiral. This "hold time" varies according to the instantaneous angular position of the drum relative to the crank at the moment of entry of the object into the machine. If :

n = number of turns in the spiral

t_{\min} = minimum hold time

t_{\max} = maximum hold time

t_{mean} = $(t_{\max} + t_{\min})/2$

$t_{\min} = \frac{n}{2\pi\omega_2}$ seconds

$t_{\max} = \frac{n+1}{2\pi\omega_2}$ seconds.

A model eccentric spiral machine was built, and used to demonstrate the principle of operation to interested persons. For reasons explained in Section II.4 this model was not useful for predicting the performance of a full size machine for centrifuging sheepskins.

According to the criteria stated in Section I.5 this eccentric spiral system appeared to have more desirable characteristics than any other system considered previously.

CHAPTER SEVEN

DESIGN OF THE ECCENTRIC SPIRAL MACHINE

An eccentric spiral machine was designed and constructed using the existing motor, drive and support hardware. An isometric section of the machine is shown in Drg CC37.

VII.1 RELATIONSHIP BETWEEN DIMENSIONS

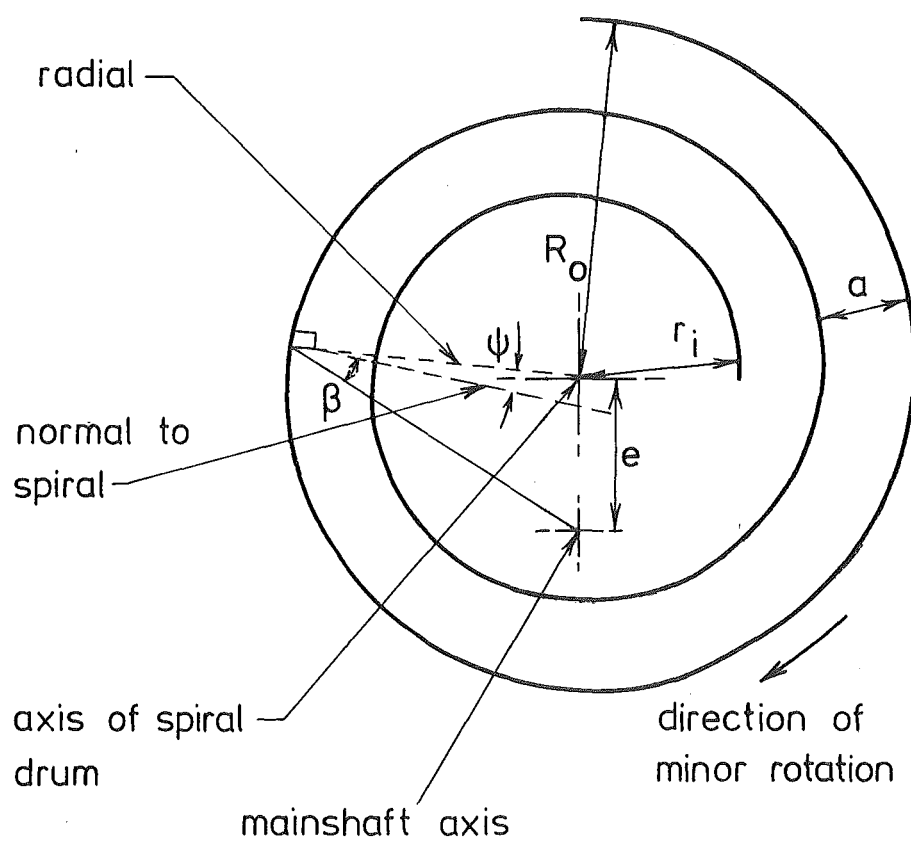
The general dimensions of the machine were influenced by three factors :

1. The total drive power required for the machine;
2. The size of openings necessary for skin movement;
3. The eccentricity necessary to ensure that the skin would slide on the spiral wall.

If, with reference to Figure 28, p. 75 :

- e = distance between mainshaft axis and spiral cylinder axis
- R_o = radius of the outer end of the spiral drum wall
- r_i = radius of the inner end of the spiral drum wall
- β = angle between the normal to the spiral wall at any point and the acceleration vector
- β_{max} = maximum value of β for a point adjacent to the outer end of the spiral wall
- P = power required to overcome aerodynamic drag
- L = continuous length of the spiral wall for which $\beta > \tan^{-1} \mu$
- μ = coefficient of friction between the object being centrifuged and the spiral drum wall
- a = radial spacing between adjacent turns of the spiral
- n = number of turns in the spiral
- r_{in} = radius of entry hole

FIG. 28. PLAN VIEW OF A $2\frac{1}{4}$ TURN CONSTANT RADIAL SPACING SPIRAL.



ψ = angle between the tangent to the spiral at a point and the normal to the line joining that point to the drum centre

$k_{\text{subscript}}$ are constants.

Then:

1. $P \approx k_{10}(R_o + e)^{\frac{1}{2}}$ (Section III.1)
2. $r_{in} = r_i - e$
3. $a = f(R_o, r_i, n)$
4. $\beta_{\max} = \sin^{-1}(e/R_o) + \psi$
5. $L = f(e, R_o, r_i, n).$

These relationships were used in an empirical manner to determine suitable dimensions for the machine. The dimensions shown were necessarily a compromise between conflicting requirements, for example, largest possible entry hole size and largest possible value of β_{\max} .

VII.1.1 Spiral Drum Geometry

The number of turns in the spiral affects the deviation from the mean hold time (Section VI.1)

For a two-turn spiral the range of hold time is $t_{\text{mean}} \pm t_{\text{mean}}/3$. The time spent by the skin against the outer turn of the spiral wall (where the acceleration field is greatest) is approximately constant, irrespective of the instantaneous orientation of the drum when the skin is fed into the machine. This variation in hold time was not expected to cause significant variation in the moisture content of otherwise similar skins, providing the time spent in the outer turn of the spiral was sufficient for equilibrium (Section I.3.2) to be closely approached.

A "pseudo spiral" in the form of two incomplete cylinders connected by a joining plate (Figure 29, p. 78) has advantages over other spiral forms. The ratio R_o/r_i for a two-turn spiral for constant values of L , r_{in} and a , is smaller for the cylindrical "pseudo spiral" than for other spiral forms. This means that the overall machine dimensions are smaller if a cylindrical "pseudo spiral" is used. The mechanical construction of a cylindrical "pseudo spiral" is less complicated than for other spiral forms.

For these reasons a two-turn cylindrical "pseudo spiral" form (hereinafter termed "the spiral") was used.

VII.1.2 Value of "L"

An eccentric spiral system will function correctly for small objects if $\beta_{\max} > \tan^{-1} \mu$ for all points on the spiral. The length (L) around the spiral wall for which $\beta > \tan^{-1} \mu$ affects the maximum size of object for which the machine will function correctly. The length L for a cylindrical wall, radius R , is given by*:

$$L = 2 R \cos^{-1} \left(\frac{R}{e} \cdot \frac{\mu}{\sqrt{1 + \mu^2}} \right)$$

The values of L for $\mu = .27$ (see Section VII.4) were plotted for various values of R and e , (Figure 30, p. 79).

VII.1.3 Power Consumption

The total power required to drive an eccentric spiral machine is made up of :

- a. Aerodynamic drag
- b. Kinetic energy input to the objects being centrifuged
- c. Friction between the object and the spiral
- d. Spiral drum bearing friction
- e. Drive losses.

* see Appendix F

FIG. 29. A TWO TURN CYLINDRICAL 'PSEUDO SPIRAL'.

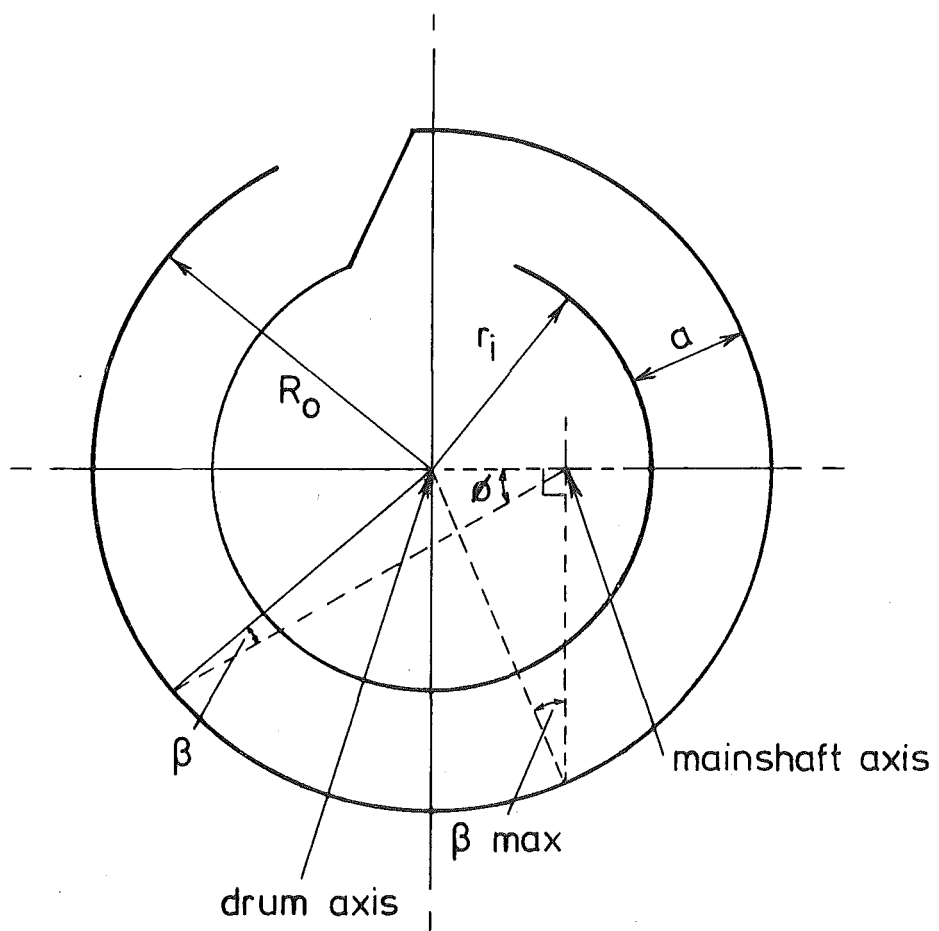
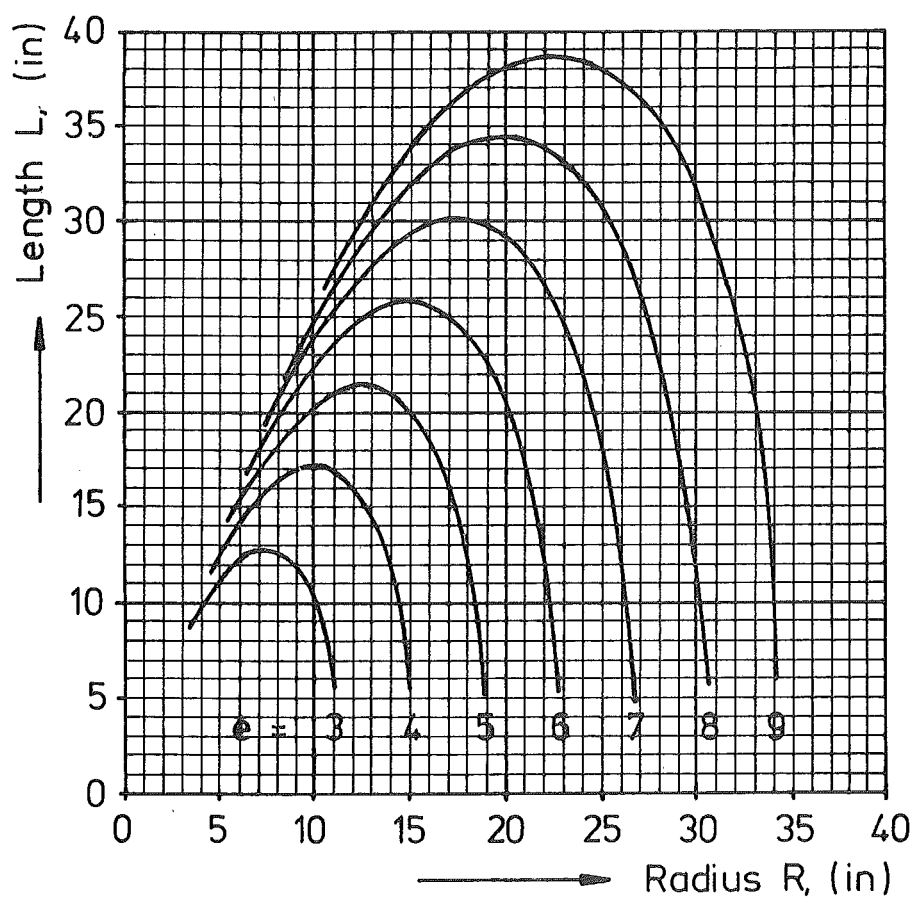


FIG. 30. VALUES OF 'L' FOR VARIOUS VALUES OF 'e' AND
'R' FOR A CYLINDRICAL 'PSEUDO SPIRAL,'
 $\mu = 0.27$.



VII.1.4 Effect of Drain Holes on Coefficient of Friction

Experiments were conducted to determine the effect of drain holes on the coefficient of friction between wool and low carbon steel plate. Two plates were prepared, one with a 5/16" diameter hole, and one with a shrouded 1/2" diameter hole of similar form to that used on the spiral, drawing CC34a. A piece of sheepskin was attached across the end of a 9 in. diameter open ended cylinder (Figure 31, p. 81). The cylinder was filled with sand to give an interface pressure during tests of approximately 1 p.s.i. The force required to drag the loaded skin along the plate was measured.

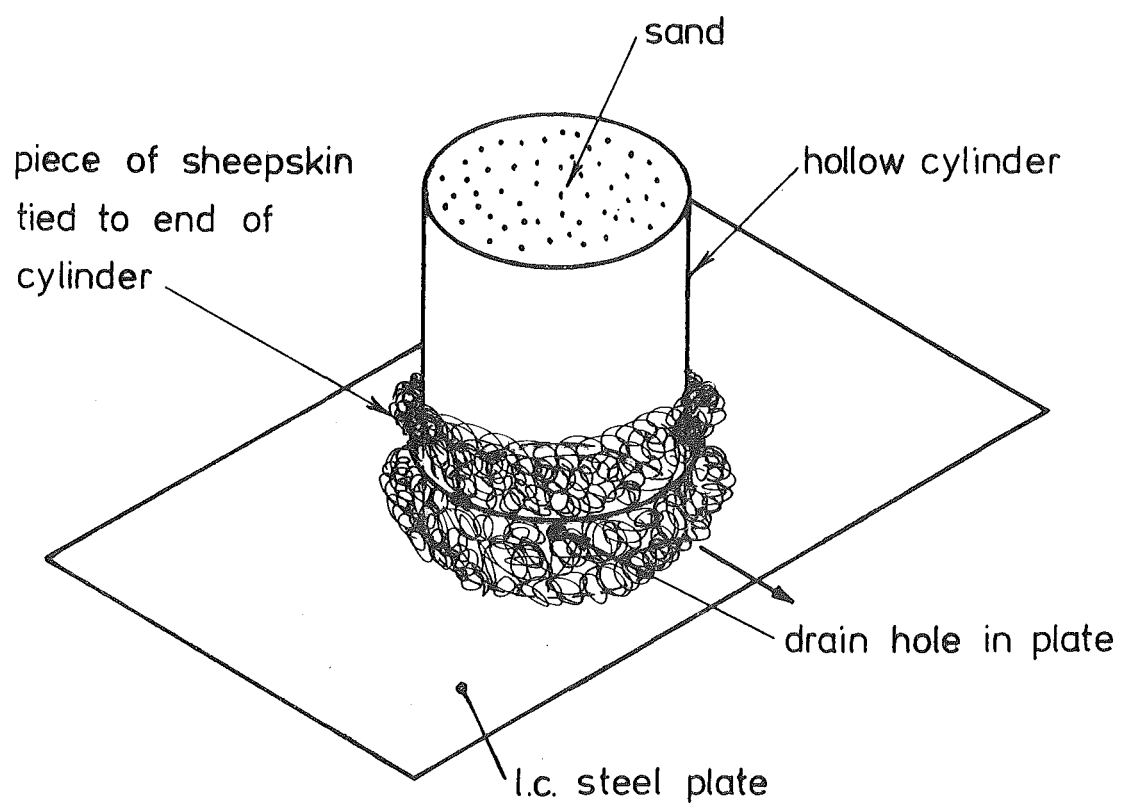
It was found that neither the 5/16" diameter hole nor the shrouded 1/2" diameter hole increased the force by more than five percent over that required using undrilled plate.

These experiments showed that at interface pressures of approximately 1 p.s.i. the effective coefficient of friction between sheepskin wool side down and the low carbon steel spiral would be approximately .27.

VII.1.5 Feed Hole Diameter

The diameter of the central feed has a maximum value of $2(r_1 - e)$. Experiments further to those described in Section III.1 were conducted involving passing wet skins through a circular hole with a conical entry. It was found that all the skins available at the time (up to 35 lbs wet weight) would pass through a 10" diameter hole as a result of their own weight, providing the skin was not bunched up immediately prior to entry.

FIG. 31. FRICTION TESTS.



VII.1.6 Drum Height and Spiral Spacing

Experiments with the frustum machine had shown that a 30lb (wet weight) sheepskin dropped into a circular rotating drum would spread out along the drum wall to a radial thickness of less than six inches, when confined within a six inch region in the direction parallel to the drum axis. This gave an approximate indication of the cross-sectional area required for the annulus between the spiral walls in the eccentric spiral machine.

VII.2 MACHINE DIMENSIONS

The following machine dimensions were selected using the relationships given earlier in this chapter to determine the effect of varying geometry within the limitations imposed by skin size and available drive power :

R_o = radius of the outer turn of the spiral
= 19"

r_i = radius of the inner turn of the spiral
= 12"

e = distance between drum axis and mainshaft axis
= 7"

H = drum height (i.e. axial dimension)
= 9".

Thus :

L (outer turn) = 30" for $\mu = .27$

L (inner turn) = 26.7" for $\mu = .27$

r_{in} = 5"

P_{tot} \approx 24 h.p.

VII.3 DRAIN HOLES

Drain holes were required to allow the escape of water from the spiral. Accurate prediction of a suitable drain hole configuration was not possible, because neither the rate of water loss at any one time during the centrifuging process, nor the quantitative effect of the woolly skin partly restricting some drain holes was known.

It was desirable both for ease of construction, and to minimise any increase in the coefficient of friction between the skin and the spiral wall, to use the minimum satisfactory number of drain holes. It was reasoned that increasing the number of drain holes beyond a certain number would have negligible effect on wool moisture content (Section I.3.4).

Drain holes were initially spaced at slightly wider intervals than an approximate prediction based on orifice flow calculations, with the intention of increasing their number after a series of test runs and comparing wool moisture contents detained using the two different configurations.

Drain holes were of two types, and were provided only on the outer turn of the spiral (Drawing CC 34a). Plain unshrouded holes were made in the light gauge steel tubing at the junction of the outer turn of the spiral and the joining plate, to discharge the initial volume of water leading the skin into the annulus. A similar row of holes was made in the tubing at the exit end of the outer turn. Six shrouded holes were equally spaced around the outer turn.

VII.4 SUPPORT OF THE SPIRAL DRUM

The spiral drum was supported on the rotating hub assembly both by a central shaft and by teflon-composite pads

(Item 136, Drg CC28) at the ends of the spars. Sufficient clearance was allowed in the bronze bearing (Item 113, Drg CC28) for the pads to sustain loading which would otherwise result in a high bending moment on the shaft. The base plate of the spiral drum was also relieved of this bending loading allowing the use of thinner sections in the spiral drum than would otherwise have been necessary.

VII.5 SPIRAL DRUM DRIVE

The drive to the spiral drum was transmitted by means of a two stage chain reduction because of the ease of alteration of ratio and the lower cost compared with alternative systems. The primary driver sprocket was keyed to one end of a shaft passing through the centre of the mainshaft, the other end of this shaft being anchored to the base frame (Drg CC37).

VII.6 COUNTERBALANCE

A fabricated counterbalance structure was fitted to the rotating assembly (Item 126, Drg CC31,32). This was designed to give dynamic balance when a skin weighing 20 lbs was in the annulus farthest from the main centre of rotation. The peak out of balance loading on the base frame was thus effectively halved when centrifuging skins weighing 40 lbs.

VII.7 ENTRY FEED GUIDE

A stationary feed guide was attached to the top cover of the machine coaxial with the mainshaft. This guide was made up of a conical frustum top section and a cylindrical lower section which protruded approximately 1" into the spiral drum.

VII.8 INITIAL VALUES OF ω_1 AND ω_2

The existing belt drive to the mainshaft was used initially, giving a mainshaft speed of 380 r.p.m. (39.8 rad/sec). This resulted in an acceleration field of 100g at the point on the outer turn of the spiral farthest from the mainshaft axis.

Although this acceleration field was lower than that predicted necessary by Rands (140g), experiments at this speed were considered worthwhile in view of the saving in power consumption possible if satisfactory water removal could be achieved.

The ratio of the chain drive to the spiral drum was initially selected to give $t_{\min} = 6.8$ seconds ($\omega_2 = .92$ rad/sec.)

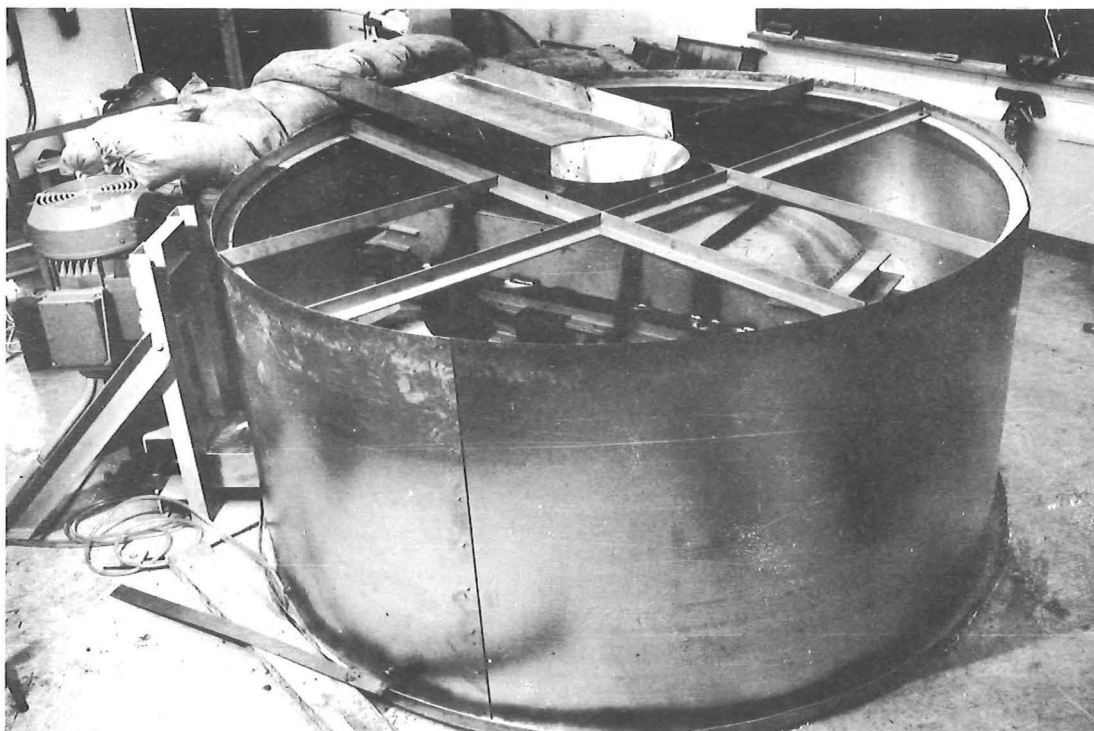


PLATE 2. THE FIRST PADDLE GATED MACHINE.



PLATE 3. VIEW FROM ABOVE OF THE FIRST
ECCENTRIC SPIRAL MACHINE.

CHAPTER EIGHT

OPERATION OF THE ECCENTRIC SPIRAL MACHINE

The machine was built in the form shown in Drg CC37.

The machine was first run without sheepskins, and a test for critical speed performed similar to that described in Chapter Four. No critical speed was found. Test runs were then made with wet woolly sheepskins.

Operation of the mechanical handling system of the machine was initially unreliable. Sections VIII.1 and VIII.2 describe the malfunctions, and modifications undertaken to remedy them.

VIII.1 PREMATURE SKIN ESCAPE

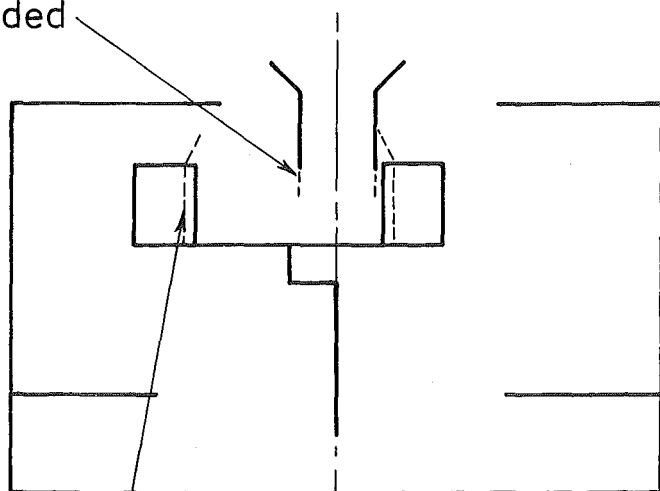
Some skins dropped into the machine rode up the inner turn of the spiral and escaped over the top of the drum, instead of entering the annulus as required.

Experiments in which the air flow through the spiral was modified by flat plates over or adjacent to the exit end showed that this air flow contributed to premature escape but was not the sole cause. It was apparent that the stiffness of the skins was sufficiently low to allow fluid-like flow under the influence of the acceleration field towards the open top of the inner turn of the spiral. The escape of a small part of the skin then dragged the remainder with it.

The feed guide cylinder was lengthened by 3" (Figure 32, p. 88), leaving a gap of 5" between the base of the drum and the end of the feed guide. This reduced the proportion of prematurely escaping skins from approximately 30% to approximately 20% of those fed into the machine.

FIG. 32. MODIFICATIONS TO PREVENT PREMATURE
SKIN ESCAPE (SHOWN THUS ----)

feed guide cylinder
extended



inner spiral radius increased 2"
and conical frustum attached.

The radius of the inner turn of the spiral was increased by 2" and a conical frustum fitted to the top of the inner turn (Figure 32, p. 88). This modification had the desired effect; no skins subsequently put through the machine escaped over the top of the inner spiral turn. The reduction in radial spacing of the spiral turns (dimension "a") from 7" to 5" did not affect skin movement through the machine.

VIII.2 SKIN ADHESION TO THE SPIRAL

Some skins did not slide against the outer spiral wall, and were carried around the drum centre by the rotation of the drum relative to the crank. This modification occurred most frequently for the first few skins put through the machine after it had been left idle for several days. Oxidation of the steel spiral, chemical change in the wool/grease and suint deposited on the spiral from previous runs, or a combination of these increased the coefficient of friction between the sheepskin and the spiral sufficiently to impede operation.

Either a reduction in the coefficient of friction between the skin and spiral wall, or an increase in $\frac{\text{eccentricity}}{\text{maximum spiral radius}}$ (with a consequent increase in L_{inner} , L_{outer} and β_{max}) could have given satisfactory operation. Increasing the value of e/R_0 while maintaining the entry guide diameter at 10" could only be achieved by an increase in machine dimensions. Modification of the surface of the spiral was preferable.

The coefficients of friction between low carbon steel and wet wool, and galvanised steel and wet wool were compared by experiment at interface pressures between 1 p.s.i. and 5 p.s.i. The figures were similar for the clean surfaces, but the galvanised surface consistently exhibited a slightly

lower value (3% to 10%, depending on the individual skin) after both test plates had been left for several days after the first tests.

The entire drum was hot dip galvanised, both as protection for the steel against corrosion, and to determine whether the resulting reduction in μ was sufficient to give reliable machine operation.

Test runs with the galvanised drum still showed somewhat unreliable machine operation.

Coefficient of friction experiments were performed for wet wool against a number of non-metallic materials, including polythene, polyamide, polytetrafluoroethylene and a molybdenum disulphide impregnated polyamide ("Nylatron GS"). All these polymers exhibited a lower (5% - 10%) coefficient of friction than galvanised steel against wet wool. "Nylatron GS" was available in strip from 6" wide and 1/16" thick, and is often used as a bearing material for water lubricated journal or linear bearings. The suppliers claimed "excellent abrasion resistance" for this material.

The inner surface of the outer turn of the spiral was lined with "Nylatron GS". The lining was secured at the leading edge with "pop" rivets and the remainder bonded with contact adhesive. This surface was found to be satisfactory, no further problems being experienced with failure of the skins to slide on the spiral.

After several weeks use some small areas of the "Nylatron" lining were lifting from the spiral. From the appearance of the areas in which bond failure was occurring it was suspected that the "Nylatron" had expanded since fitting.

VIII.3 WATER REMOVAL

With the initial drain hole configuration and centrifuging parameters ($\omega_1 = 39.8$ rad/sec, $t_{\min} = 6.8$ sec) the wool moisture content of skins centrifuged in the eccentric spiral machine was significantly higher than that required by industry. Changes were made in ω_1 , t_{\min} , exit water control hardware and the drain hole configuration.

Table 2 (p. 92) shows the changes made, and the range of wool moisture contents obtained with the different configurations for skins with wool lengths between 2" and 6". Sections VIII.3.1 to VIII.3.3 inclusive discuss the results shown on this table in terms of the effect of varying a single machine characteristic.

VIII.3.1 Drain Hole Configuration

All the drain hole configurations used gave similar wool moisture levels for otherwise similar centrifuging parameters, although the total area of drain holes varied by greater than a factor of two. It was concluded that none of the configurations of drain holes used significantly restricted the escape of water from the skin.

The initial drain hole configuration used a combination of shrouded and unshrouded holes. Neither of these types became permanently blocked with wool fibres, so only the plain unshrouded holes were used in the latter stages of experimentation.

VIII.3.2 Discharge Water Separation

Initially there was no provision for spatial separation of water and sheepskins discharged from the machine, separation depending on the time difference between water escape and skin release. This resulted in some re-wetting of the skin as

TABLE 2

WOOL MOISTURE RANGE FOR VARIOUS MACHINE CONDITIONS

DRAIN HOLES	DISCHARGE SEPARATION	MAINSHAFT SPEED (r.p.m.)	t _{min} (seconds)	W.M.C. %
1. 6x½"φ shrouded + 8 x 5/16"φ (Drg CC34a)	None	380	6.8	55-70
2. As above plus 3 vertical rows of six 5/16"φ	None	380	6.8	55-70
3. 11 diag. rows each 6 x 5/16"φ (Drg CC38)	None	380	6.8	55-70
4. "	None	470	5.5	55-65
5. "	See arrange- ment in Fig. 38	470	5.5	50-60
6. As above plus 10 diag. rows each 6 x 5/16" φ	"	470	5.5	50-60
7. "	"	470	11	45-50
8. "	"	380	13.6	55-70
9. "	"	380	13.6 +13.6	55-70
10. "	"	470	11+11	40-50

it slid around the wet machine outer casing after discharge.

A conical frustum casing was later fitted which enclosed the drum. A gap was left in the frustum at the spiral exit. This directed water escaping through the drain holes upward, above the plane of skin exit. A stationary gutter was fitted to the centrifuge surrounding structure to collect this water and discharge it to a drain (Figure 33, p. 94).

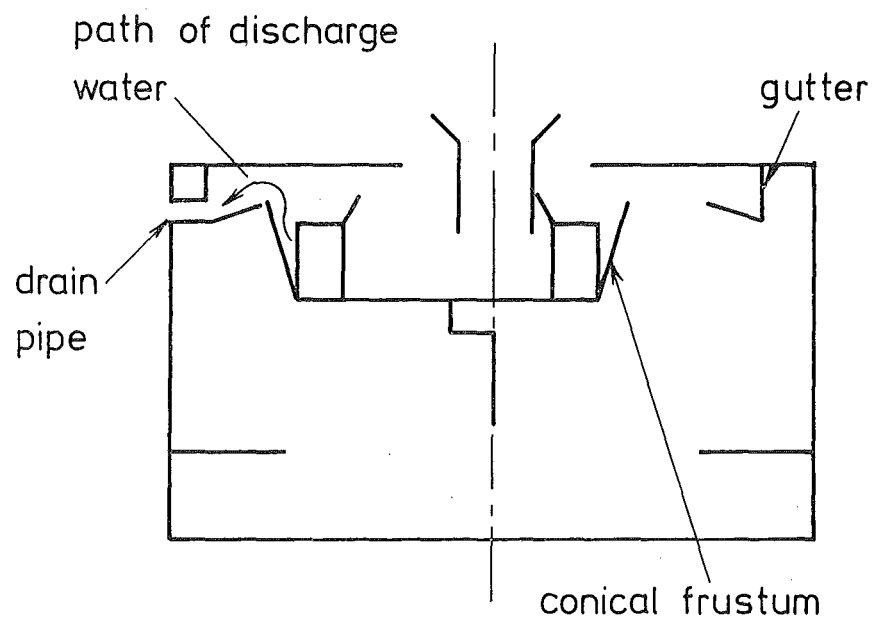
Although this addition reduced wool moisture content for otherwise identical conditions (see Table 2, 4 and 5, p. 92), separation of exit water and skin was not complete; the induced wind resulting from the rotation of the machine carried small droplets of water from the top of the water guide frustum to the centrifuge outer casing.

VIII.3.3 Mainshaft Speed and Hold Time

With a mainshaft speed of 380 r.p.m. the average wool moisture content of any one skin was constant to within $\pm 3\%$ for t_{\min} between 6.8 seconds and 13.6 seconds. Passing a skin through the machine a second time when $t_{\min} = 13.6$ seconds gave no significant further reduction in wool moisture content. It was concluded that equilibrium (Section I.3.2) was closely approached in 6.8 seconds at this mainshaft speed. The wool moisture content of skins centrifuged at 380 r.p.m. was higher than that considered acceptable by the industry.

The mainshaft speed was increased to 470 r.p.m. The wool moisture content of skins centrifuged at this speed was significantly lower for $t_{\min} = 11$ seconds than for $t_{\min} = 5.5$ seconds (see Table 2, 6 and 7, p. 92). Passing a skin through the machine a second time for $t_{\min} = 11$ seconds gave a further reduction in wool moisture content (see Table 2, 7 and 10) of typically 4% to 5%. Passing a skin through

FIG. 33. DISCHARGE WATER GUIDES.



the machine a third time for $t_{\min} = 11$ seconds gave negligible further reduction in wool moisture, usually less than 1%.

With a mainshaft speed of 470 r.p.m. and a mean hold time of 20 seconds, acceptable (45% or below) wool moisture levels could be obtained for skins carrying wool longer than 2".

The wool moisture content for skins with less than $1\frac{1}{2}$ " of wool was usually greater than 45%.

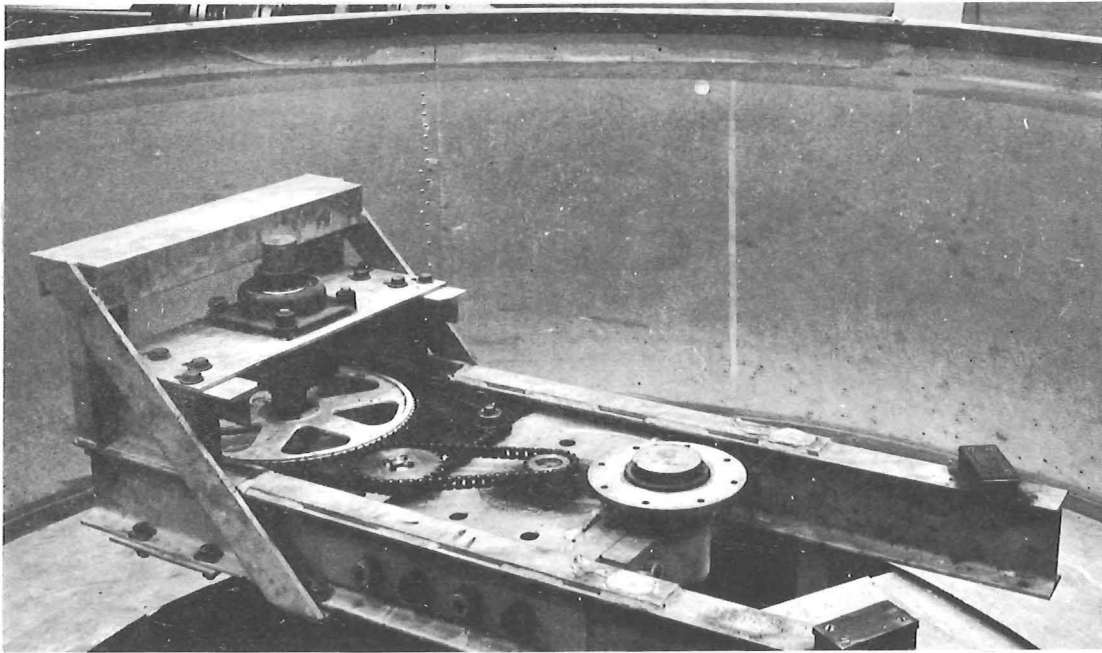


PLATE 4. THE ECCENTRIC SPIRAL MACHINE
WITH THE TOP COVER AND SPIRAL REMOVED.

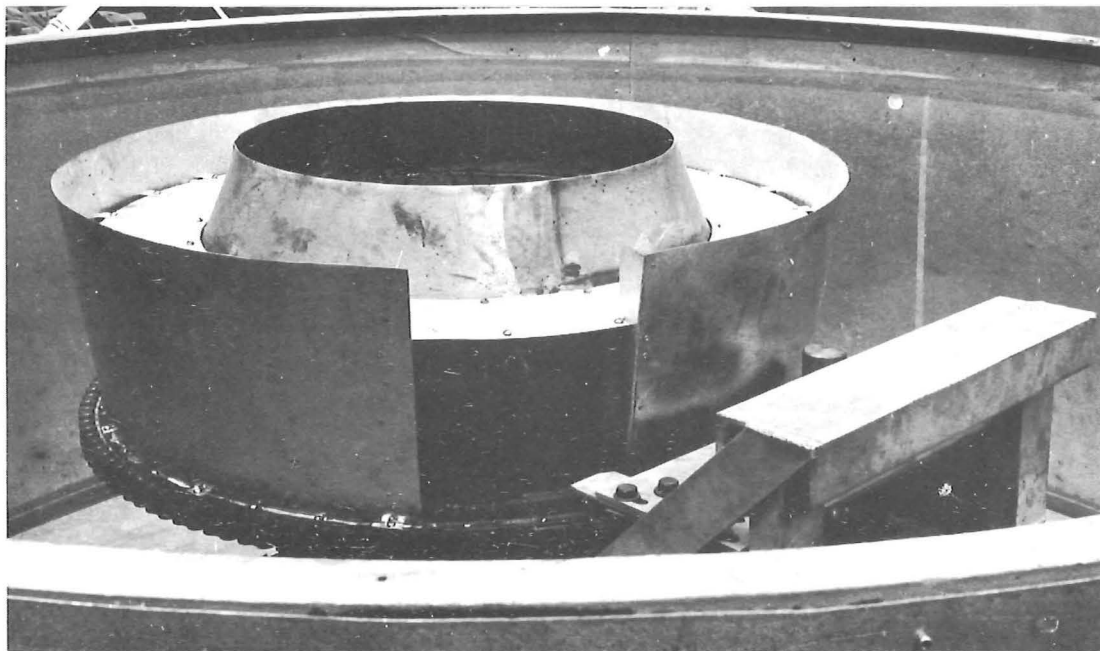


PLATE 5. THE FINAL ECCENTRIC SPIRAL MACHINE
WITH THE TOP COVER REMOVED.

CHAPTER NINE

PERFORMANCE OF THE FINAL VERSION OF THE EXPERIMENTAL ECCENTRIC SPIRAL MACHINE

IX.1 WOOL MOISTURE CONTENT

The continuous centrifuge running at 470 r.p.m. did not quite reduce the wool moisture of woolly skins to the levels achieved by conventional "hydro" extraction. It was thought that an increase in the magnitude of the centrifugal acceleration field of the continuous machine would be necessary to obtain similar wool moisture content by either process. The maximum speed of the experimental centrifuge was limited to 500 r.p.m. by stress levels in load bearing machine parts, so experiments at speeds significantly higher than 470 r.p.m. were not possible with this machine. Wool moisture contents for a number of skins, obtained both by "hydro" extraction and continuous centrifuging at 470 r.p.m. are shown in Table 3.

Continuously centrifuged skins usually had less variation in wool moisture content between different points on the skin than did "hydroed" skins. This was more pronounced for skins with more than 2" of wool than for skins with shorter wool.

Some re-wetting of the woolly skin occurred when it came into contact with the centrifuge outer casing after discharge from the rotating system (Section VIII.3.2). The small quantity of water adhering to the casing contributed to the measured wool moisture, and had greater effect on the moisture content of skins with shorter wool. This could account in part for the greater variation in wool moisture content over the area of skins with shorter wool.

AVERAGE WOOL LENGTH	EX-HYDRO AVERAGE W.M.C.	EX-HYDRO W.M.C. STD.DEV.	EX-CONT. CENTR. 1 PASS (11 SECONDS) AV. W.M.C. 470 rpm	EX-CONT. CENTR. 1 PASS (11 SECONDS) W.M.C. STD.DEV. 470 rpm	EX-CONT. CENTR. 2 PASSES (22 SEC- ONDS) AV. W.M.C. 470 rpm	EX-CONT. CENTR. 2 PASSES (22 SEC- ONDS) W.M.C. STD. DEV. 470 rpm
1"	48%	3.9%	58%	9.1%	52%	2.2%
1"	44%	6.3%	49%	8.4%	-	-
2"	44%	3.1%	48%	3.0%	45%	1.3%
3"	42%	3.06%	50%	2.04%	-	-
4"	39%	4.08%	45%	3.2%	42.9%	1.1%
4½"	40%	3.5%	43%	3.1%	42.4%	1.1%

TABLE 3

Each horizontal line is for one individual skin. Average wool moisture content and standard deviation are from six randomly chosen samples from each skin for each centrifuging condition.

The trend for increasing wool moisture with decreasing wool length could have resulted from a combination of three factors :

- (1) The different equilibrium wool moisture content for different wool lengths.
- (2) Re-wetting, as described above; and
- (3) Migration of water from the skin to the wool between centrifuging and removal of the wool for moisture content measurement.

IX.2 REPEATABILITY

The effect on moisture content of random variations in hold time (Section VII.1.1) and skin orientation was found to be small. A woolly skin with an ex-hydro weight of 13 lbs and 2½" of wool was alternately soaked and centrifuged at 470 r.p.m. for $t_{\text{mean}} = 22$ seconds. For six such cycles the ex-centrifuge weight remained within $\pm 1\frac{1}{2}\%$ of the mean value.

IX.3 SKIN DAMAGE

Woolly skins centrifuged at 470 r.p.m. in the experimental machine were examined by the New Zealand Leather and Shoe Research Association. Although damage was found in two skins, this damage was not directly attributable to the centrifuge. Other skins centrifuged were found to be of normal quality.

IX.4 SKIN MOISTURE CONTENT

Three skins centrifuged at 470 r.p.m. for $t_{\text{min}} = 22$ seconds were returned to C.F.M. (Belfast) Freezing Works .

for painting and pulling. Depilatory action was normal and no difficulty was experienced in removing the wool.

CHAPTER TEN

CONCLUSIONS

1. An experimental machine was designed and constructed which was capable of performing the mechanical handling operations required of a continuous centrifuge for woolly sheepskins.
2. The experimental machine could process woolly sheepskins without damage.
3. The experimental machine could reduce the wool moisture content of woolly sheepskins to within 5% w.m.c. of those obtained by conventional "hydro" extraction. Theory (Section I.3.2) suggests that an increase in the rotational speed of the experimental machine could result in similar w.m.c. from either centrifuging process.
4. Throughput capability of the experimental machine was approximately three skins per minute. No reasons were known precluding satisfactory operation of a larger machine processing several skins simultaneously, thus having a higher throughput capability.
5. A commercial machine operating on the principle developed would not require an operator.
6. A machine operating on the principle developed could be of rugged construction without delicate components or mechanisms requiring frequent re-setting to compensate for wear.
7. Periodic adjustment of centrifuging parameters (hold time and acceleration field) to allow for seasonal variations in woolly sheepskin characteristics could be achieved by altering drive ratios (Section VI.1).

8. The sole restriction on skin feed timing with this system is a limit on total mass input during any one period of twice the maximum hold time. (see Section VI.1).

CHAPTER ELEVEN

FUTURE DEVELOPMENT

It is considered that a successful commercial machine could be designed operating on the eccentric spiral principle developed with the experimental machine.

Apart from the obvious additions of a feed conveyor belt and a discharge conveyor belt system, a commercial machine could advantageously differ from the experimental machine in a number of respects.

A machine suitable for handling all sizes of skin would be larger than the experimental machine. Approximately 25% increase in linear dimensions would be satisfactory.

The spiral could be lined with stainless steel rather than "Nylaton". Both the greater dimensional stability of stainless steel and the lower coefficient of friction between sheepskin and stainless steel than sheepskin and "Nylaton" would be advantageous.

A cylindrical fairing coaxial with the mainshaft would reduce aerodynamic drag and induced wind, thus decreasing both power consumption and wetting of the outer collecting wall by water spray.

A reduction in the radial clearance between the rotating parts and the collecting cylinder wall would result in lower impact force on the collecting cylinder and reduce any possibility of skin damage.

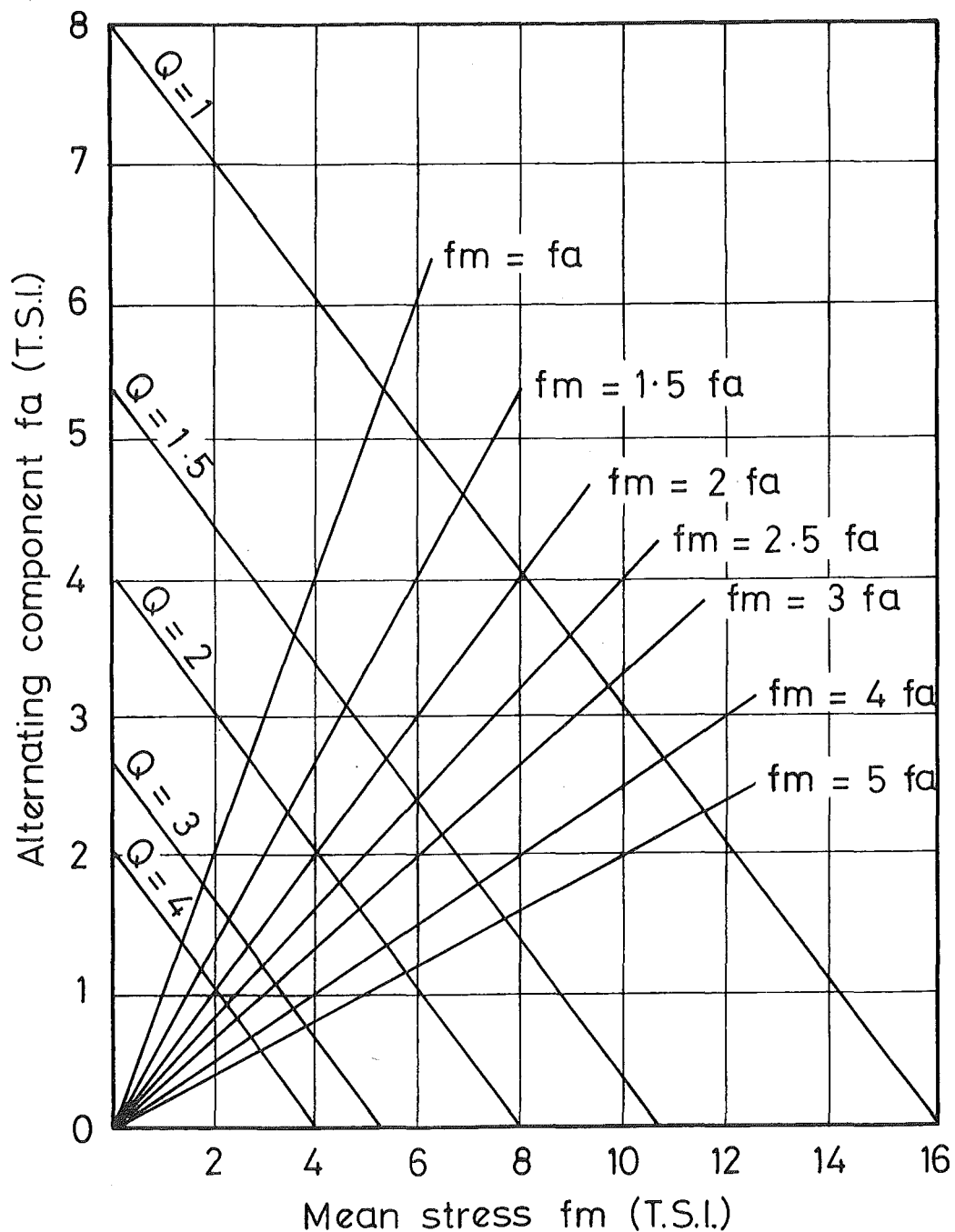


Fig 34 SODERBERG DIAGRAM FOR LOW CARBON STEEL TO BS15 IN WATER. f_a MAX. FROM REF. 5. f_m MAX. FROM BS15.

APPENDIX A

STRESS CALCULATIONS

All values for Q (safety factor) in this Appendix are taken from Fig.16, the Soderberg diagram for the material used.

The following abbreviations are used :-

- A = load bearing area
- B.M. = bending moment
- I = second moment of area
- K = stress concentration factor
- M = moment
- P = tensile or compressive load
- Q = safety factor
- SF = shear force
- U.D.L.= uniformly distributed load
- y = distance from neutral axis to extreme fibre
- σ = tensile stress
- σ_x, σ_y = principal stresses
- τ = shear stress

A.1 THE PADDLE GATED CENTRIFUGE

The loading on many parts of the machine could be divided into three contributing parts, the constant loading resulting from the centrifugal acceleration of the machine components, the loading resulting from the centrifugal acceleration of a skin or skins in the machine, and the loading resulting from impact of a skin against a paddle blade. Of these three, only impact loading could not be accurately predicted.

An estimate of impact loading was essential to the structural design of the machine. Ignoring friction forces in the radial direction, it was shown that a point mass dropped into the machine running at 380 r.p.m. would have a velocity v at radius r (ft) :

$$\begin{aligned} v &= 38.6r \\ &= 64.5 \text{ f.p.s.} \end{aligned}$$

on striking the paddle blade.

It was considered that friction would reduce this figure to less than 50 f.p.s. It was assumed that the C.G. of the skin moved at least 4" between arrest of the leading part of the skin and arrest of the trailing part of the skin. Using these figures, the impact force :-

$$\begin{aligned} F &= v^2 / (2S) \\ &= 5850 \text{ lbs.} \end{aligned}$$

Effective weight of the skin at the paddle axis radius is 3850 lbs.

These two figures are not directly additive to give peak loading on the paddle blade. A maximum figure of 7000 lbs was assumed for force exerted by a skin on a vertical paddle blade.

(i) Paddle Wheel:

"Stresscoat" tests indicated that for 13.4 tons load, $\sigma_{\max} = 14$ t.s.i. for test quadrant constructed from $\frac{3}{16}$ " plate. For $\frac{1}{8}$ " plate and 7000 lbs load assuming stress inversely proportional to thickness

$$\sigma_{\max} = 4.9 \text{ t.s.i.}$$

$$\sigma = 2.45 \pm 2.45 \text{ t.s.i.}$$

$$Q \approx 2$$

(ii) Paddle Shaft (DRG.CC16):

Bending stress at mid point:

Worst case loading - 50 lb skin striking vertical paddle blade.

$$\text{Skin load} = 7000 \text{ lbs}$$

$$\text{Effective weight of paddle wheel} = 6320 \text{ lbs}$$

$$\text{Effective shaft UDL} = 200 \text{ lbs/inch}$$

Thus max. BM in shaft 36,500 lb.in which for $3\frac{1}{2}$ " diameter shaft gives:

$$\sigma = \pm 3.87 \text{ t.s.i.}$$

$$Q \approx 2$$

Shear stress adjacent to bearing:

If skin load centre of effort is 4" from trough wall,

$$\text{S.F.} = 7910 \pm 2800 \text{ lbs}$$

$$\tau = 1600 \pm 570 \text{ p.s.i.}$$

$$Q > 4$$

(iii) Paddle Shaft Bearing Housing to Spar Weld (DRG.CC16):

$$\text{Load} = 7910 \pm 2800 + 410 \text{ lbs}$$

$$A = 4\frac{1}{4} \text{ in}^2$$

$$\text{if } K = 4$$

$$\sigma = 3.5 \pm 1.2 \text{ t.s.i.}$$

$$Q \approx 2$$

(iv) Spars (DRG.CC16):

$$A = 4\frac{1}{4} \text{ in}^2$$

$$\text{load} = 8300 \pm 2800 \text{ lbs}$$

$$\text{if } K = 3 \text{ at bolt holes}$$

$$\sigma = 2.7 \pm .9 \text{ t.s.i.}$$

$$Q \doteq 3$$

(v) Trough Ends (DRG.CC18):

$$\text{Load from skin} = 7000 \text{ lbs.}$$

Assuming this is distributed in a manner equivalent to two superimposed U.D.L.s of 280 lbs/in., one over the entire width of the trough and another acting only over the central 6", then BM at mid point = 17,600 lb/in.

$$\text{Mass of trough end including ribs} = 30 \text{ lbs.}$$

$$\text{Total BM} = 30,000 \text{ lb/in.}$$

$$\text{Thus } \sigma_{\text{bending}} = 7100 \pm 2900 \text{ lbs}$$

$$Q \doteq 3.$$

(vi) Trough Walls (DRG.CC18):

For 50 lb skin in each end:

$$\text{Load} = 11,200 \text{ lbs}$$

$$\text{over } 42" \times .104"$$

$$\text{gives } \sigma = 1.2 \text{ t.s.i.}$$

$$= .6 \text{ t.s.i.} \pm .6 \text{ t.s.i.}$$

$$Q > 4$$

(vii) Trough Wall to Spar Attachment (DRG.CC18):

14g wall to 2" x 2" x $\frac{1}{4}$ " angle weld.

$$\text{Effective weld area} = 7.2 \text{ in}^2 \text{ per side.}$$

$$\text{If max. load} = \frac{2}{3} \times 7000 \text{ lbs/side}$$

$$\text{and } K = 4$$

$$\tau = \pm 1.2 \text{ t.s.i.}$$

$$Q > 3$$

Similarly for bolts securing this angle to the spar:

$$\tau = \pm 3.9 \text{ t.s.i.}$$

$$Q = 1\frac{1}{4} \text{ neglecting friction.}$$

(viii) Bolts Securing Spars to Main Hub:

Eight 1" BSW "Unbrako" capscrews, U.T.S. approx.
90 t.s.i.

Maximum loading occurs with 50 lb skin in one end
and a second 50 lb skin striking the vertical
paddle blade in the same end of the trough.

$$\begin{aligned} \text{Max. load} &= 10,300 \text{ lbs per side} \\ &= 2.1 \text{ t.s.i. neglecting friction} \end{aligned}$$

Shear stress in bolts resulting from loading being
applied in a plane above that of the bolts:

$$BM = 72,100 \text{ lbs}$$

Ignoring inner two bolts:

$$\tau = 5.85 \text{ t.s.i.}$$

$$Q > 7 \text{ on U.T.S.}$$

(ix) Hub (DRG.CC5):

For tensile load of 7000 lbs in each spar maximum
stress in hub:

$$= 935 \text{ p.s.i.}$$

$$Q > 5$$

(x) Mainshaft (DRG.CC4):

Highest stress is at the top of the top bearing,
i.e. 6.375" from the top of the shaft.

For one 50 lb skin in one end of the trough and a
second striking the vertical paddle blade shaft:

$$BM = 133,000 \text{ lbs/ins}$$

$$K = 1.8 \quad (5)$$

$$\sigma = \pm 3.95 \text{ t.s.i.}$$

Power transmitted through this shaft, 25 h.p. at 380 r.p.m. results in shear stress in this region of

$$\tau = 75 \text{ p.s.i.}$$

Shear load from out of balance:

$$= 14,000 \text{ lbs}$$

results in:

$$\tau_2 = 425 \text{ p.s.i.}$$

$$Q \doteq 2$$

(xi) Main Bearings Support Structure (DRGS CC7 & CC.9):

Effective maximum load is equivalent to 14,000 lbs in a plane $9\frac{1}{2}$ " above the top of the top main bearing, thus load on the top bearing = 30,000 lbs

$$\text{load on the bottom bearing} = 16,000 \text{ lbs.}$$

Shear stress in legs immediately below top bearing housing is 30,000 lbs over 9.8 in^2

$$= 1.4 \text{ t.s.i.}$$

$$Q > 3$$

Region in legs immediately above lower bearing housing is subject to shear and tensile/compression loading.

$$\text{Shear Load} = 30,000 \text{ lbs}$$

$$\text{Area} = 4 \times 3.8 \text{ in}^2$$

$$\tau = 2000 \text{ p.s.i.}$$

Distance between centroids of opposite legs

$$= 17.7"$$

If $K = 3$ at the weld

$$\sigma = 9600 \text{ p.s.i.}$$

Continuing τ and σ gives principal stresses:

$$\sigma_x = 10,000 \text{ p.s.i.}$$

$$\sigma_y = -400 \text{ p.s.i.}$$

Using the Total Shear Strain Energy theory,

$$\text{Equivalent stress} = \pm 4.5 \text{ t.s.i.}$$

$$Q > 1.5$$

Similarly in the region immediately below the lower bearing housing

$$\sigma_{\text{equiv}} = \pm 4 \text{ t.s.i.}$$

$$Q > 1.5$$

and in the legs immediately above the feet:

$$\sigma_{\text{equiv}} = \pm 4.8 \text{ t.s.i.}$$

$$Q > 1.5$$

(xii) Base Frame (DRG.CC.12):

Peak out of balance load is 14,000 lbs 27" above the centre of base frame.

Legs are affixed to frame on corners of a 24" square.

∴ Maximum load in each leg = 7900 lbs

Thus BM in base frame immediately below leg attachment points = 63,000 lbs/in.

$$I = 19.5 \text{ in}^4.$$

Thus $\sigma_1 = 8000 \text{ p.s.i.}$

Direct stress resulting from out of balance force is 14,000 lbs over 15 in^2 .

$$\sigma_2 = 900 \text{ p.s.i.}$$

thus $\sigma = \pm 4 \text{ t.s.i.}$

$$Q = 2.$$

(xiii) Floor Connections:

Load per floor socket = 5,400 lbs

$$= 7600 \text{ p.s.i. in } \frac{3}{4} \text{ bolts}$$

acceptable for h.t. bolts.

(xiv) Mainshaft bearings were designed to withstand 30,000 lbs and 16,000 lbs loading respectively, and give satisfactory service life for industrial machinery according to the SKF bearing manufacturers' specifications.

(xv) Transmission Components (belt drive, centrifugal clutch, primary shaft bearings), were selected in accordance with manufacturers' specifications for the transmission of 25 h.p.

A.II THE FRUSTUM CENTRIFUGE(i) Tyre Characteristics Required:

For 50 lb skin in 100g acceleration field spread
over 2 ft of the tyre circumference,

$$\text{tyre ID} = 32\frac{1}{2}" , \text{O.D.} = 39\frac{1}{2}" .$$

Pressure exerted by skin on tyre

$$= 30 \text{ p.s.i.}$$

\therefore Tyre pressure should be 30 p.s.i. to prevent
release.

Treating the tyre as a straight thin wall cylinder,
the wall loading is:

$$P_{\text{axial}} = 50 \text{ lbs/hoop inch}$$

$$P_{\text{hoop}} = 100 \text{ lbs/axial inch.}$$

Additional load imposed by effective weight of the
skin $\hat{=}$ 70 lbs/axial inch.

Thus tyre material should be capable of withstanding

$$P_{\text{axial}} = 50 \text{ lbs/inch}$$

$$P_{\text{hoop}} = 170 \text{ lbs/inch.}$$

- (ii) Loading on other components is similar to or less
than as described for the paddle gated trough machine.

A.III THE ECCENTRIC SPIRAL CENTRIFUGE

Mass of spiral drum (DRG.CC34a) = 200 lbs.

At 7.26" radius and 500 r.p.m. apparent weight
= 10,000 lbs.

At maximum radius the apparent weight of a 50 lb skin
is 9420 lbs.

∴ Total load = 20,000 lbs, 5" above plane of bottom
of spiral drum.

(i) Bolts (Item 108, DRG.CC28):

Total load = 6470 lbs

for K = 3

$\sigma = 11.3 \text{ t.s.i.}$, satisfactory for
"Unbrako" capscrews.

(ii) Spiral Drum Shaft (DRG.CC33):

$\tau = 4150 \text{ p.s.i.}$

for K = 3

$\sigma = 3950 \text{ p.s.i.}$

∴ $\sigma_x = 2625 \text{ p.s.i.}$

$\sigma_y = 6575 \text{ p.s.i.}$

From Total Strain Energy Theory

$\sigma_{\text{equiv}} = 2.8 \text{ t.s.i.}$

$Q > 4$

Integral shaft cap:

P = 6470 lbs

$A_{\text{shear}} = 4.9 \text{ in}^2$

∴ $\tau = .59 \text{ t.s.i.}$

$Q > 4$

(iii) Clamp (DRG.CC33):

Shear failure under capscrew heads:

$$\text{Load} = 20,000 \text{ lbs}$$

$$\tau = .67 \text{ t.s.i.}$$

$$Q > 4$$

Tensile Failure:

$$A = 6 \text{ in}^2$$

$$\sigma = 1.5 \text{ t.s.i.}$$

$$Q > 4$$

(iv) Capscrews Securing Clamp to Hub:

Four 1" x 2½" BSW "Unbrako" type:

$$\text{Load} = 20,000 \text{ lbs}$$

$$\text{for } K = 3$$

$$\sigma = 12 \text{ t.s.i.}$$

(v) Pads (Item 132, DRG.CC35):

$$\text{Load} = 6470 \text{ lbs}$$

If maximum load per pad = 4000 lbs, then maximum

bearing load on "Glacier DU" pads = 666 p.s.i.

which is satisfactory for this material.

(vi) Spars:

B.M. resulting from pad loading of 4000 lbs

$$= 100,000 \text{ inch/lbs.}$$

$$I \text{ of composite spar} = 22 \text{ in}^4$$

$$\sigma_{\text{bending}} = 5.8 \text{ t.s.i.}$$

$$= 2.9 \pm 2.9 \text{ t.s.i.}$$

$$Q = 2$$

The counterbalance structure was designed such that the machine was dynamically balanced with one 20 lb skin in the outer part of the spiral at the maximum radius.

- (vii) Chain drive was designed to transmit 10 h.p. with a safety factor > 10 on the breaking load of the chains. This would not give suitable chain and sprocket life in a commercial machine, but was satisfactory for the experimental machine.
- (viii) Bronze bushes and thrust washers used on the spiral drum shaft were designed for 4000 p.s.i. projected area bearing pressure.
- (ix) Loading on components between the hub (Item 2m) and the building floor was similar to the case of the paddle and trough machine.

APPENDIX BSTRAIN MEASUREMENT SYSTEMSI. "STRESSCOAT" BRITTLE LACQUER

"Stresscoat" ST1205 brittle lacquer was used in obtaining a load/maximum stress relationship on a test quadrant of similar geometry to the paddle blades used in the first experimental machine.

The advantages of using brittle lacquer over other systems of strain measurement for this test are its omnidirectionality in the plane of the surface and the ease with which a large surface may be monitored during loading to determine the region of maximum strain. Quantitative results obtained by this method can be influenced by lacquer age, thickness of application, humidity and temperature during drying and testing, and rate of strain. Hounsfield tensometer test pieces used for calibration were lacquered, dried and strained at the same time and under similar conditions to the test quadrant in an effort to minimise differences in lacquer behaviour.

B.II STRAIN GAUGE TELEMETRY

An "Aerotherm" strain telemetry system was used for externally monitoring the output of strain gauge bridges affixed to rotating parts of the machine.

Slip rings were considered initially, but rejected because of the problems involved in reducing electrical noise to an acceptable level in the adverse environment.

The system comprised a Static Strain Transmitter, Model 206 and a Model 140s receiver fitted with a suitable signal conditioning card.

The system was particularly useful for measuring the effect of impact loading of a skin striking a vertical paddle blade on the strain in the rotating structural parts of the machine. Calculations indicated that a rise time of about 6 msec. could be expected for the superimposed strain pulse remitting from impact. The rise time of the telemetry system is specified at .4 msec.

Errors in the system are specified at $\pm 2\%$ of maximum strain for the range in use, corresponding to approximately ± 300 p.s.i. stress for 500 μ strain f.s.d.

A typical arrangement used would comprise: a full bridge of 350 Ω foil gauges feeding the telemetry transmitter; the received signal after decoding in the receiver would be directly fed to an oscilloscope, giving a time-strain analogue trace on the x and y axes respectively.

The system was calibrated statically using a Hounsfield tensometer and test piece and a Hounsfield extensometer. Strain versus output linearity was found to be accurate within the limits of the measuring oscilloscope for this static testing.

APPENDIX C

MEASUREMENT OF WOOL MOISTURE

Two systems of measurement of retained water are used in the wool industry.

"Regain", defined as $\frac{\text{weight of retained water}}{\text{weight of dry wool}}$ is traditionally used at lower wool moisture levels.

"Wool Moisture Content", the parameter generally used in this thesis is defined as $\frac{\text{weight of retained water}}{\text{weight of wet wool}}$.

Two methods are available to determine the effect on wool moisture content of modification to centrifuging machinery, viz :-

1. Processing a statistically significant sample of skins both before and after the modification and comparing the average wool moisture content for each group.

2. Processing one or more particular skins both before and after the modification, and comparing the average wool moisture content for each skin centrifuged under the two conditions.

For economic reasons the latter method was used during the work described.

When a wool moisture measurement for a woolly skin was required, six samples of approximately 20 gms weight were clipped off close to the skin and weighed. After drying for 45 minutes at 110 C in a temperature controlled forced air vent oven, the samples were reweighed and the moisture content calculated.

Statistical theory shows (assuming normal distribution) that for an average wool moisture content of 50% and a

standard deviation over the skin of 4%, 93% of randomly selected groups of six samples from the skin will have an average wool moisture content within 3% of the true average over the skin.

APPENDIX D

THEORETICAL AERODYNAMIC POWER PREDICTION FOR THE PADDLE-GATED MACHINE

Reference 4 gives drag coefficients for various flat plates at angles of attack to a fluid flow between 0° and 90° . With angles of attack between 45° and 90° when the flow is separated, the drag coefficient for the normal force is 1.0 to 1.3.

It was assumed that small areas of the upstream faces of the trough would have approximately the same drag coefficient as an inclined plate with an angle of attack equal to the angle between the trough wall at the centre of the area and the instantaneous direction of motion of the area (see Figure 35, p. 122).

It was assumed that the normal drag coefficient was 1.3 irrespective of radius. Although the angle of attack is less than 45° at radii less than 9 in., these areas make a very small contribution towards the total aerodynamic drag power.

The trough was treated as a rectangular box, and the protruding paddle blades as rectangular boxes with one face normal to the air flow (Figure 36, p.123).

(a) Trough:

$$r = .75/\sin\theta_a$$

$$\therefore v = .75\omega/\sin\theta_a = (.75 \times 39.8)/\sin\theta_a$$

$$\Delta\ell = (.75\Delta\theta_a)/(\sin^2\theta_a) \text{ for small } \Delta\theta_a.$$

If ΔA is the area of the strip width $\Delta\ell$

$$\Delta A = (1.25)(.75\Delta\theta)/(\sin^2\theta_a).$$

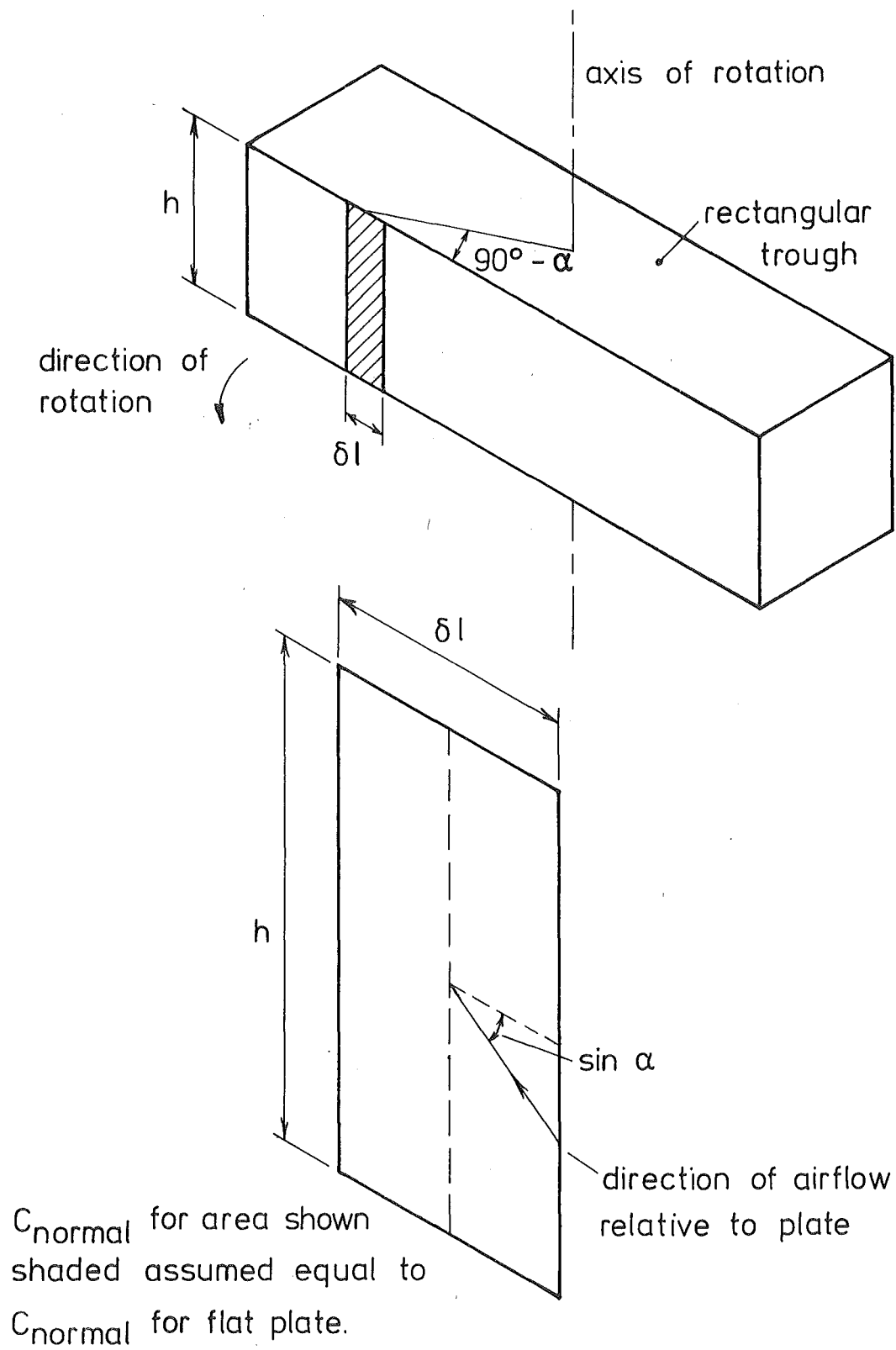
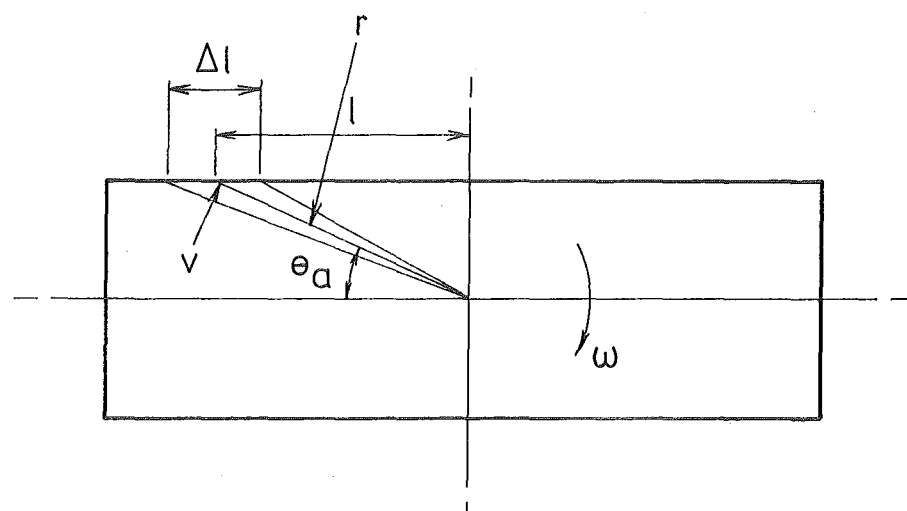
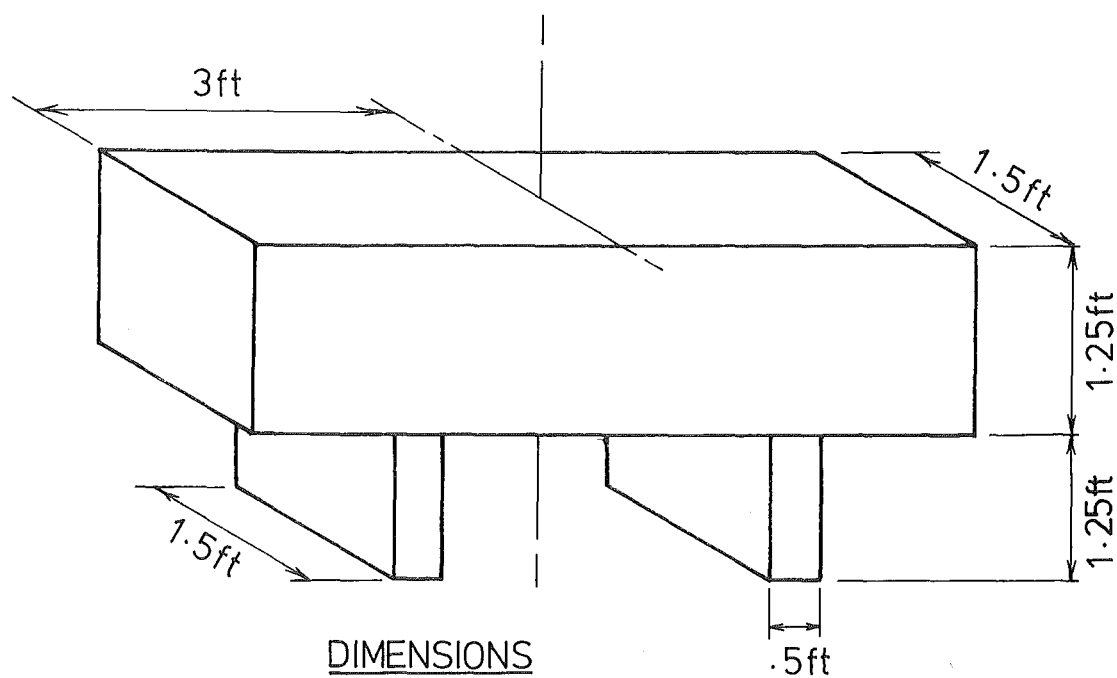


FIG. 35. AERODYNAMIC DRAG APPROXIMATION

FIG. 36. IDEALISED TROUGH AND PADDLE GEOMETRY.



SYMBOLS (PLAN VIEW)

If ΔF is the aerodynamic force on the element ΔA

$$\Delta F_{\text{normal}} = \frac{1}{2} \rho \cdot C_n \cdot \Delta A \cdot v^2$$

$$\Delta F_{\text{tangential}} = \Delta F_{\text{normal}} \cdot \cos \theta_a$$

If ΔT is the aerodynamic drag torque about the axis of rotation, then

$$\Delta T = \Delta F_{\text{normal}} \cdot \cos \theta_a \cdot r$$

Substituting:

$$\Delta T = .00125 \times 1.3 \times \frac{(1.25)(.75\Delta\theta)}{\sin^2 \theta_a} \times \frac{(.75 \times 39.8)^2}{\sin^2 \theta_a}$$

$$\times \frac{.75 \cos \theta_a}{\sin \theta_a}$$

$$T = \frac{1.02 \cos \theta_a \cdot \Delta\theta}{\sin^5 \theta_a}$$

$$\therefore \text{Total torque} = 2 \times 1.02 \int_{\theta_a = .245}^{\theta_a = \frac{\pi}{2}} \frac{\cos \theta_a}{\sin^5 \theta_a} d\theta_a$$

This integral was evaluated approximately by programming a calculator to perform the summation in .05 radian steps, and found to be 105.

Thus the total trough drag torque was predicted to be 214 lbs.ft.

(b) Paddle Blades:

The protruding paddle blades were treated as 15" x 6" rectangles normal to the air flow with a C_D of 2 and a radius of 21". Thus, the paddle drag torque about the main centre of rotation was predicted to be 27 lbs.ft.

$$\therefore \text{Total aerodynamic drag torque} = 241 \text{ lbs.ft}$$

which at 380 r.p.m.

= 17.5 hp.

MODEL STUDY

A 1/12 scale model of the rotating trough and paddle wheel assembly was constructed. The main shaft was supported by small rolling element bearings, with its axis horizontal. A torque of .0136 lb.in. was applied to the shaft by a weight attached to a length of cotton wound around the shaft. The terminal rotational speed of the model with this torque applied was measured stroboscopically to be 353 r.p.m. Bearing friction was .0003 lb.in.

It can be shown that for rotating structures of similar geometry, aerodynamic drag torque is proportional to $L^5 \omega^2$

where L = linear dimension ratio (=12)

ω = angular velocity ratio (= 1.076)

The power required to overcome aerodynamic drag was predicted from these figures to be 18 hp.

APPENDIX E

SUPPORT OF THE MACHINE

1. STIFFNESS OF THE MAIN BASE FRAME

The main base frame consisted of two pairs of 5" x 2½" channels (Drg CC12) with ¼" plate gussets. The frame was secured to the floor at eight points by means of bolts screwed into threaded sockets integral with the floor structure. The form of loading applied to the base frame is shown in Figure 37, p.127.

It was necessary to determine the deflection per unit radial load of point 0 (through which the out of balance force acts), as a result of strain in the structure.

It was shown that for small deflections the stiffness of the main base frame is constant irrespective of the direction of the force through 0, (provided the force is perpendicular to the axis of rotation). Calculations were then made, assuming this force was parallel to one of the pairs of channels, as shown in Figure 38, p. 128.

The deflection of 0 (δZ_1) resulting from bending of one pair of channels without gussets was then calculated for a radial load P at 0. As a result of ignoring the gussets and the effect of the second pair of channels, the figure for stiffness obtained would be below the true stiffness.

Triangle OBC is approximately equilateral, and for small values of δy may be considered to rotate about A.

∴ For small δZ_1 ,

$$\delta Z_1 = \sqrt{3} \cdot \delta y .$$

FIG. 37. LOADING ON MAIN BASE FRAME.

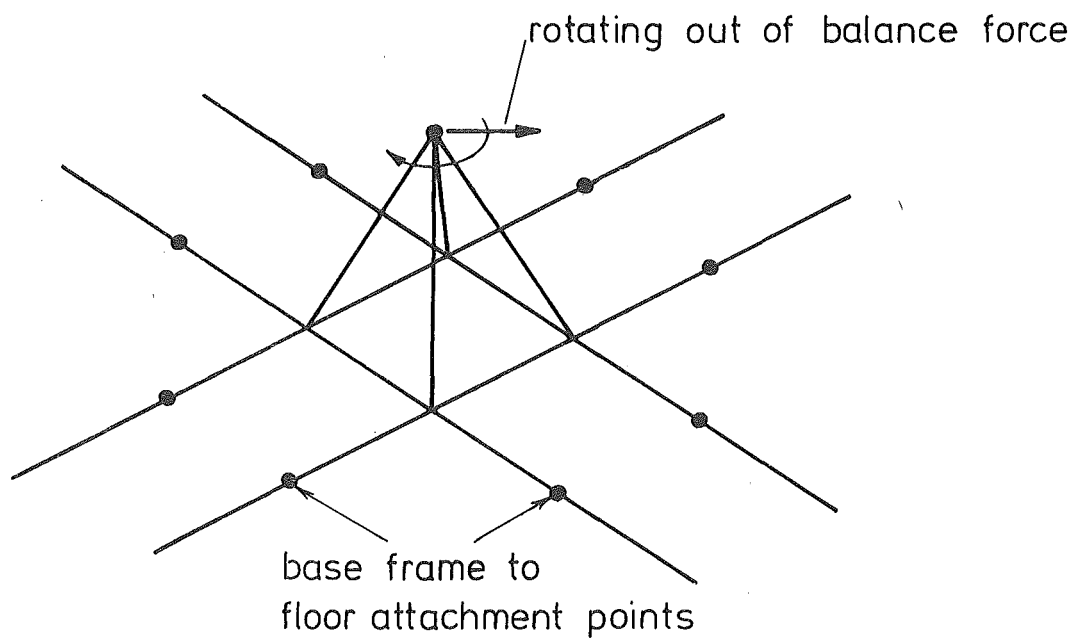


FIG. 38. BENDING OF THE BASE FRAME.

--- strained position

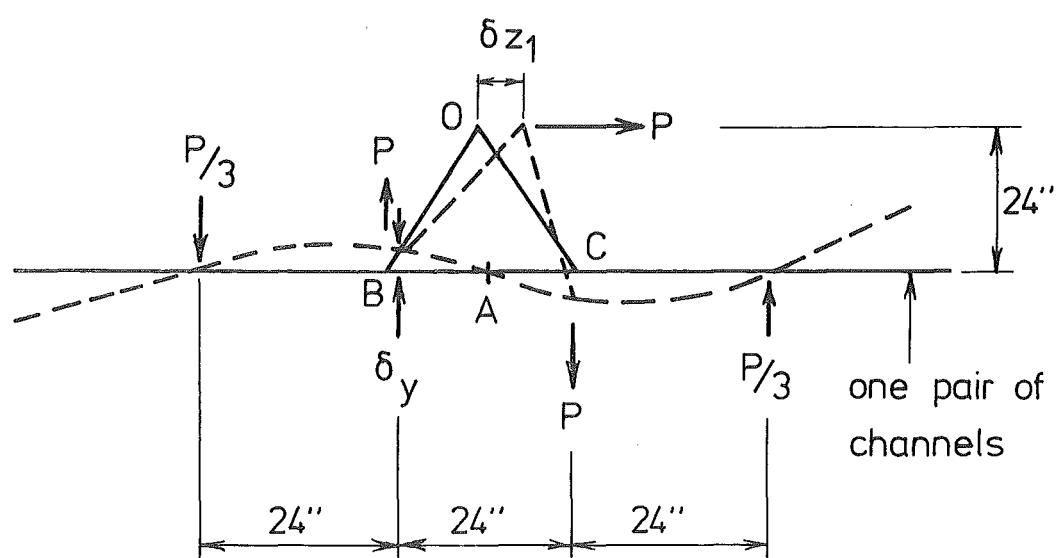
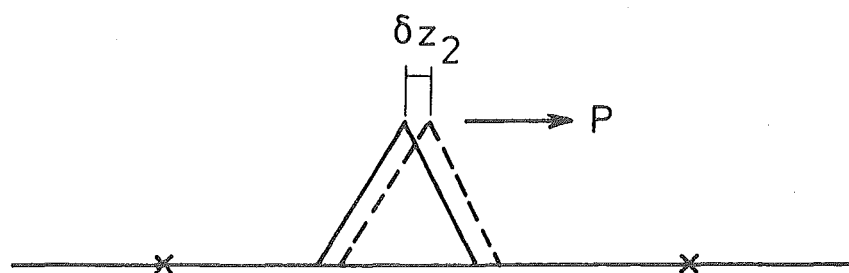


FIG. 39. STRAIN IN THE PLANE OF THE BASE FRAME.



Using MacCaulay's notation :

$$EIy'' = \frac{P}{3} x - P [x - 24] + P [x - 48]$$

Integrating twice, and introducing appropriate constants gives :

$$\frac{P}{\delta y} = 762,000 \text{ lbs/inch}$$

$$\text{i.e. } \frac{P}{\delta Z_1} = 440,000 \text{ lbs/inch}$$

The base frame is also strained in its own plane as a result of force P (Figure 39). The two channels shown in Figure 23 are axially strained, and the channel pair at right angles to these are bent.

$$\frac{P}{\delta Z_2} \text{ was shown to exceed } 2 \times 10^7 \text{ lbs/inch.}$$

2. TOTAL EFFECTIVE STIFFNESS

The effective stiffness of other parts joining the rotating structure to the building foundations was also estimated using conservative approximations. All other links in the load path had significantly higher effective stiffness than the figure obtained for the main base frame.

The total effective stiffness was in excess of 300,000 lbs per inch deflection at point 0.

3. CRITICAL SPEED

If : ω_w = min.angular velocity for whirl

g = gravitational acceleration

m = weight of rotating structure (approximately equal for paddle-gated machine and eccentric spiral machine)

$P/\delta Z$ = total effective stiffness.

then

$$\begin{aligned}\omega_w &= [(g/m) (P/z)]^{\frac{1}{2}} \\ &= [(32 \times 12/900) (300,00)]^{\frac{1}{2}} \\ &= 350 \text{ rad/sec.}\end{aligned}$$

The centrifuge design speed was 380 r.p.m. (39.8 rad/sec).

It was therefore expected that the machine would not reach critical speed.

4. LABORATORY FLOOR

The laboratory available for the machine was situated on the ground floor at one end of the Mechanical Engineering wing of the University. Plans of the building showed that this laboratory had been designed to support heavy machinery. The floor was of unusually massive reinforced concrete construction, an area approximately 8 ft by 10 ft in the centre of the room being 12" thick and the remainder 6" thick. Set into this central area was an array of threaded sockets for machine mounting.

The maximum out of balance load expected from the machine on the floor was 14,000 lbs (two 50 lb skins in a 140g acceleration field).

The advice of Dr T. Paulay of the Department of Civil Engineering was obtained on the possible consequences of loading the floor in this manner. It was concluded that failure of either the floor or the surrounding building was extremely unlikely.

APPENDIX FDERIVATION OF THE RELATIONSHIP BETWEENL, R, e and μ .

With reference to Figure 40 (p. 132) :

$\theta_1 = \theta_2$ from the definition of L, and

$\Delta_1 = \Delta_2$ from geometry.

$$\frac{R}{\sin(\Delta+\phi)} = \frac{e}{\sin\theta} = \frac{R}{\sin\phi} \quad (\text{sine rule})$$

$$\therefore \sin(\Delta+\phi) = \sin\phi \quad (1)$$

From the geometry of the figure

$$\phi < \pi/2$$

and $\Delta < \pi$.

Also, $\sin A = \sin(\pi-A)$ for any angle A.

$$\therefore \Delta + \phi = \pi - \phi, \text{ from (1)}$$

$$\therefore \phi = \pi/2 - \Delta/2 \quad (2)$$

Substituting the value for ϕ from (2) into l.h.s. of (1) and $(\frac{R}{e} \sin\theta)$ for $\sin\phi$ in r.h.s. of (1)

$$\sin(\Delta + \frac{\pi}{2} - \frac{\Delta}{2}) = \frac{R}{e} \sin\theta$$

$$\therefore \cos \frac{\Delta}{2} = \frac{R}{e} \sin\theta$$

$$\therefore \Delta = 2 \cos^{-1} \left(\frac{R}{e} \sin\theta \right) \quad (3)$$

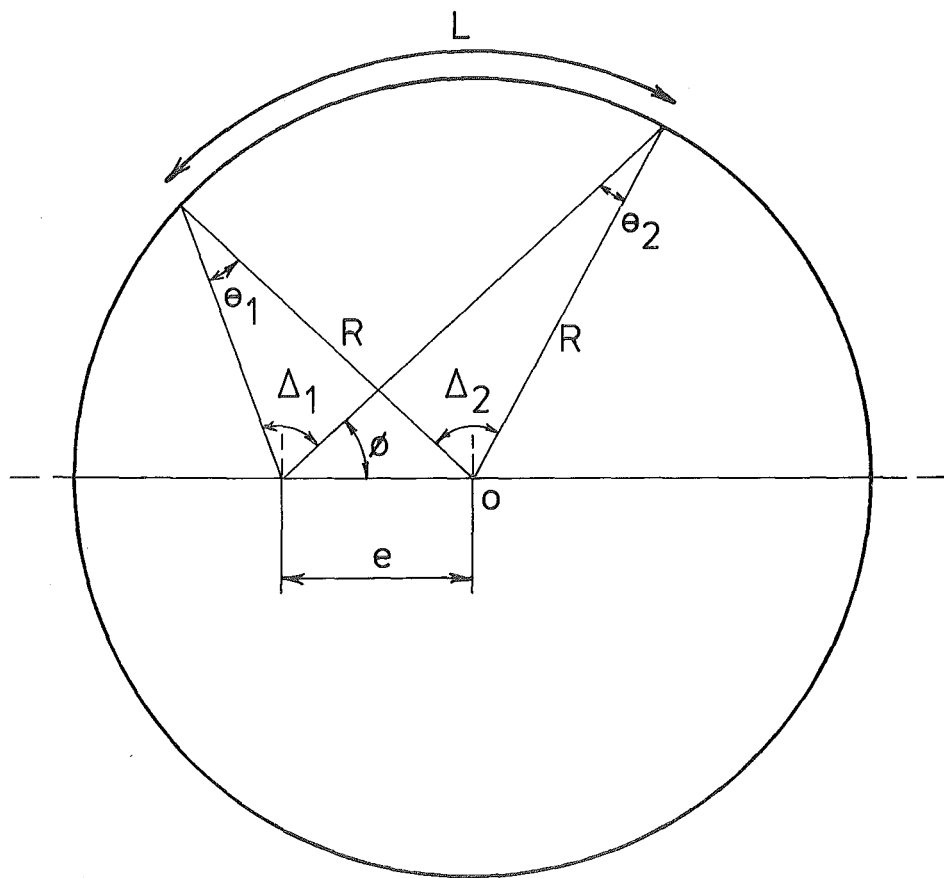
But $\theta = \tan^{-1} \mu$

$$\therefore \sin\theta = \frac{\mu}{\sqrt{1+\mu^2}}$$

Also $L = R\Delta$.

$$\therefore \text{From (3), } L = 2R \cos^{-1} \left(\frac{R}{e} \cdot \frac{\mu}{\sqrt{1+\mu^2}} \right)$$

FIG. 40. SYMBOLS USED IN THE DERIVATION OF L .



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8. Australian Wool Compendium, Australian Wool Corporation.

INDEX TO DRAWINGSA. PADDLE GATED TROUGH SERIES INCLUDING SUPPORT STRUCTURE
FOR SUBSEQUENT MACHINES

CC3	Mainshaft assembly including main bearing housing.
CC4	Mainshaft
CC5	Hub
CC6	Bottom main bearing housing
CC7	Legs
CC8	Top main bearing housing. Hub plate.
CC9	Main bearing housings and leg assembly.
CC10	Motor support assembly including primary shaft bearing structure.
CC11	Motor support structure components.
CC12	Plan view of motor support structure and base framework.
CC13	Motor support base plates.
CC14	Template for motor support base plate.
CC15	Main base framework detail.
CC16	Paddle shaft and bearing assembly. Spars.
CC17	Four bladed paddle wheels.
CC18	Trough
CC19	Floor socket plan. Outer casing.
CC20	Outer casing and other sheet steel surrounding structure.
CC21	Site plan.
CC22	Four stop cam indexing mechanism.
CC23	Trough modifications.
CC24	Three bladed paddle wheels.
CC25	Three stop cam indexing mechanism
CC26	Isometric section of the machine as first run.

B. FRUSTUM CENTRIFUGE

CC27 Frustum detail.

C. ECCENTRIC SPIRAL CENTRIFUGE

CC28 General assembly.

CC29 Hub, hub plate and mainshaft modifications.

CC30 Spiral drum transmission components.

CC31.32 Counterbalance and layshaft support structure.

CC33 Spiral drum shaft and surrounding components.

CC34a Spiral drum.

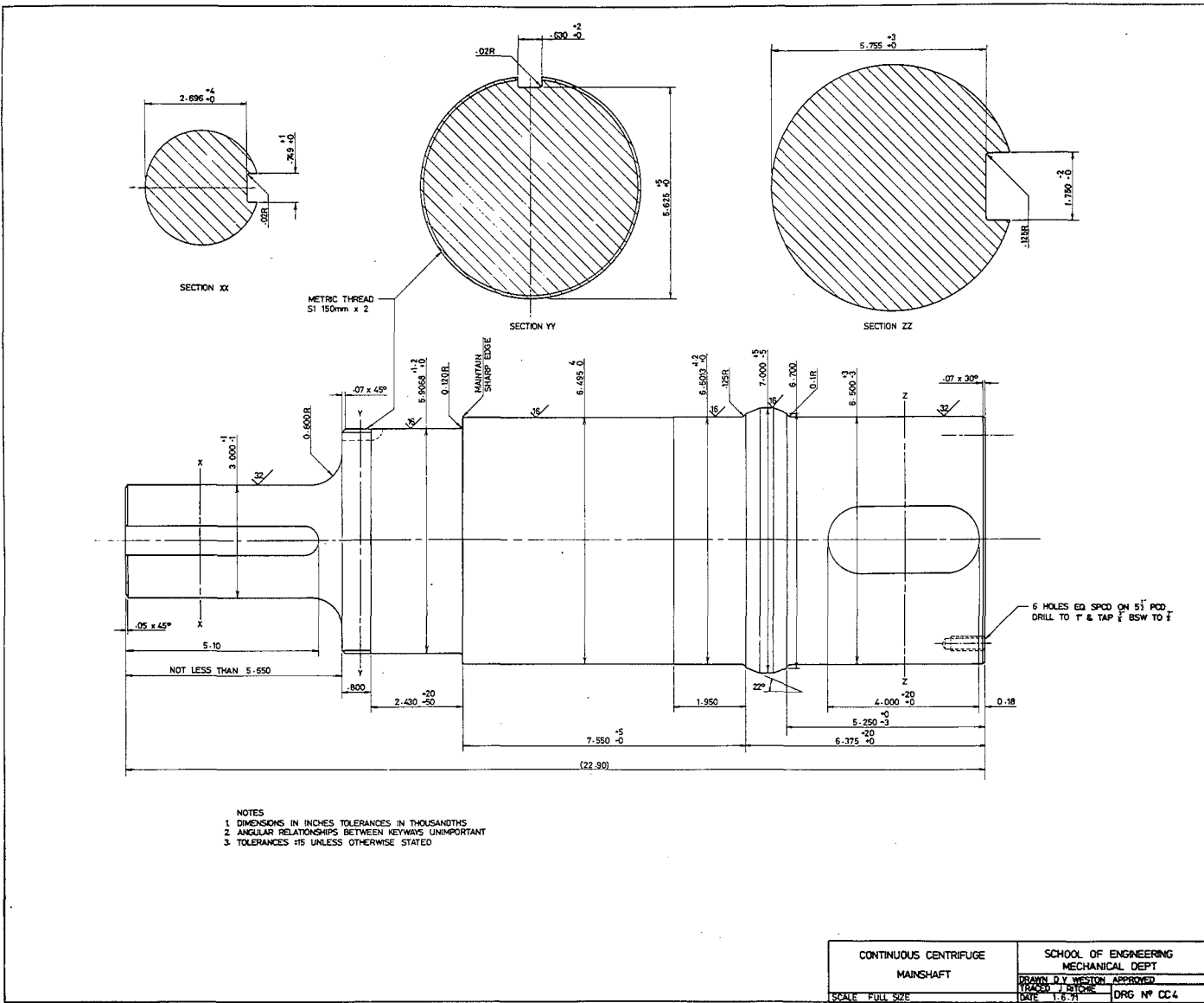
CC34b Spiral drum base plate (secondary driven sprocket).

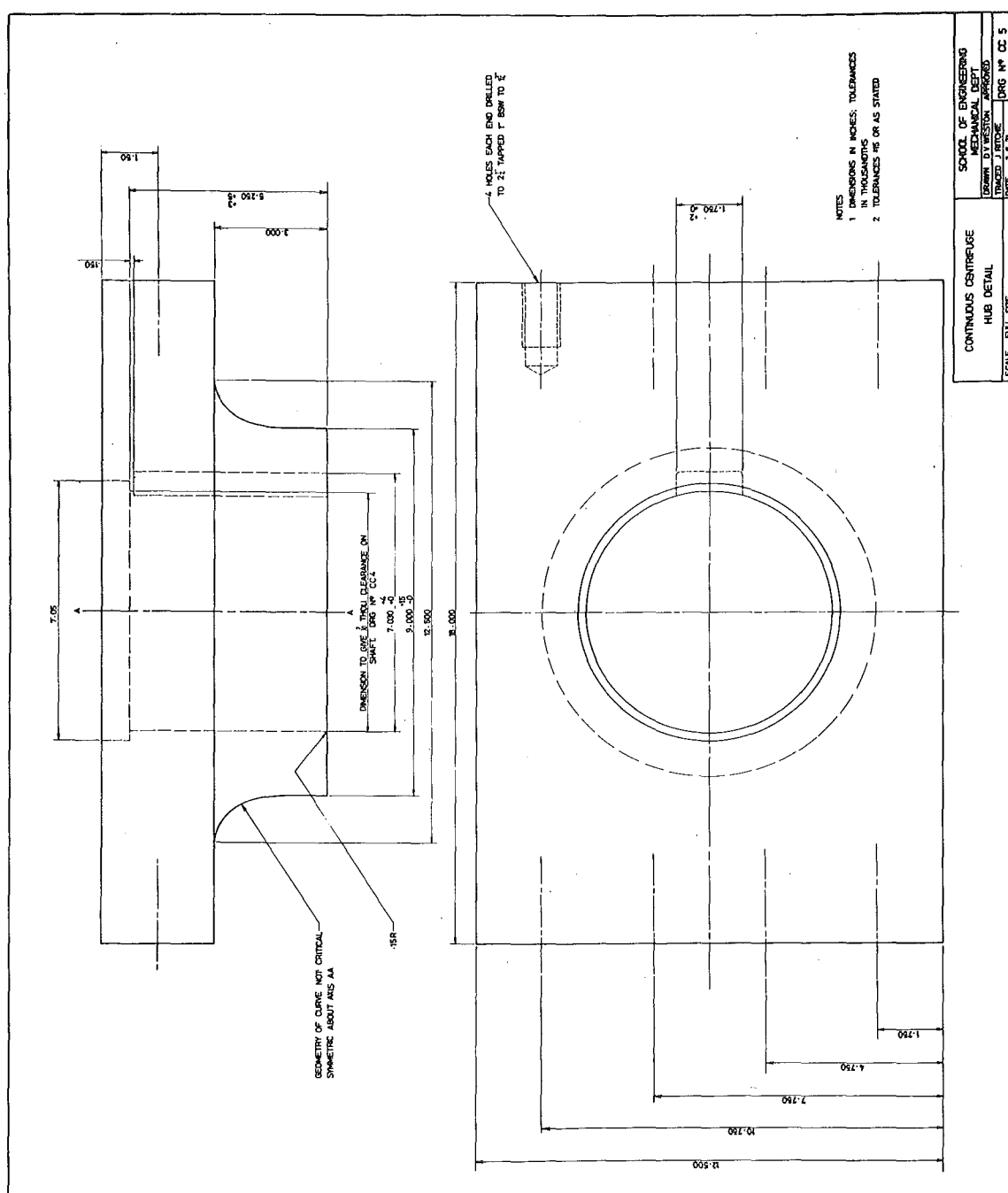
CC35 Spars.

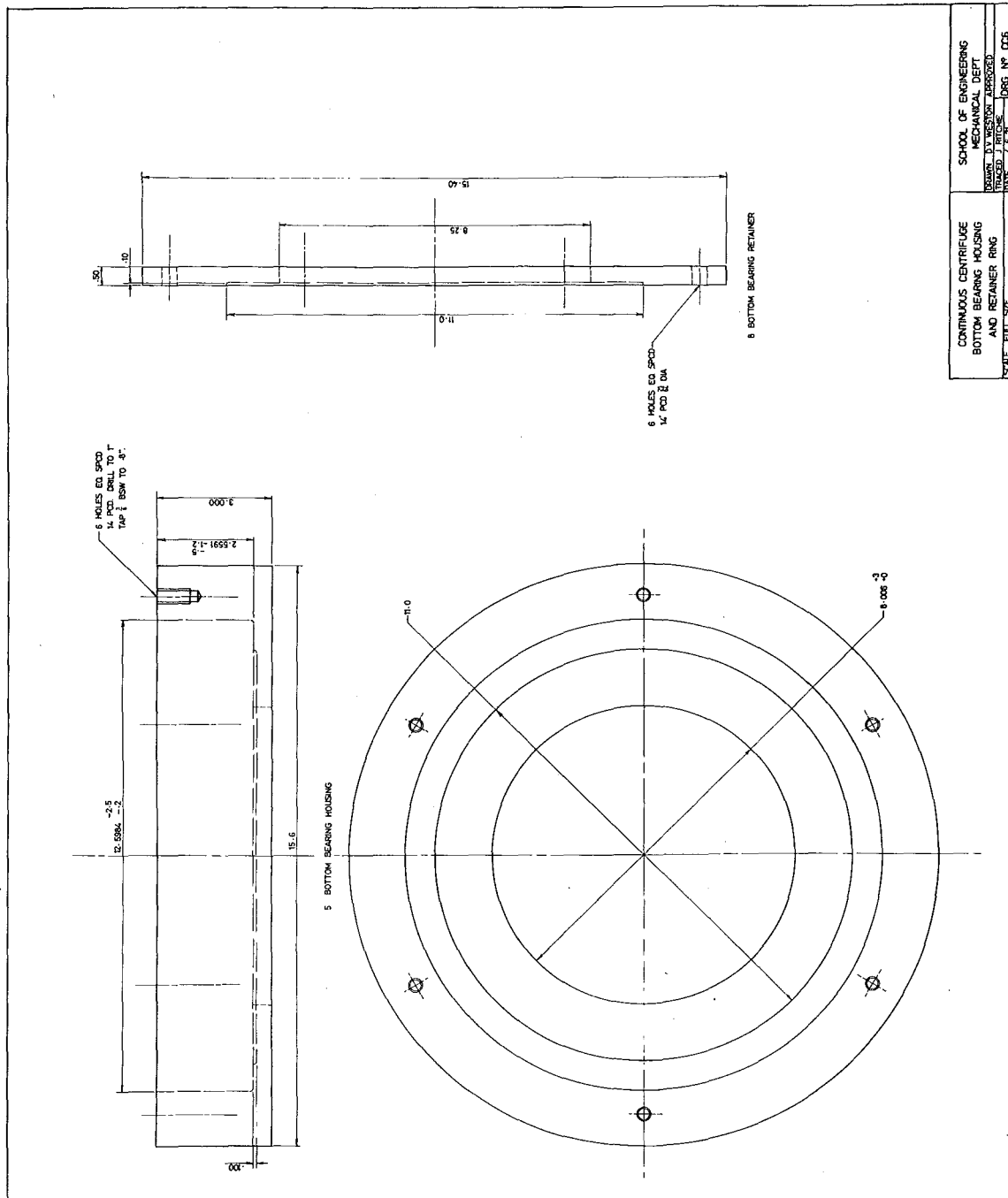
CC36 Drive shaft stay assembly.

CC37 Isometric section of the unmodified eccentric spiral
 machine.

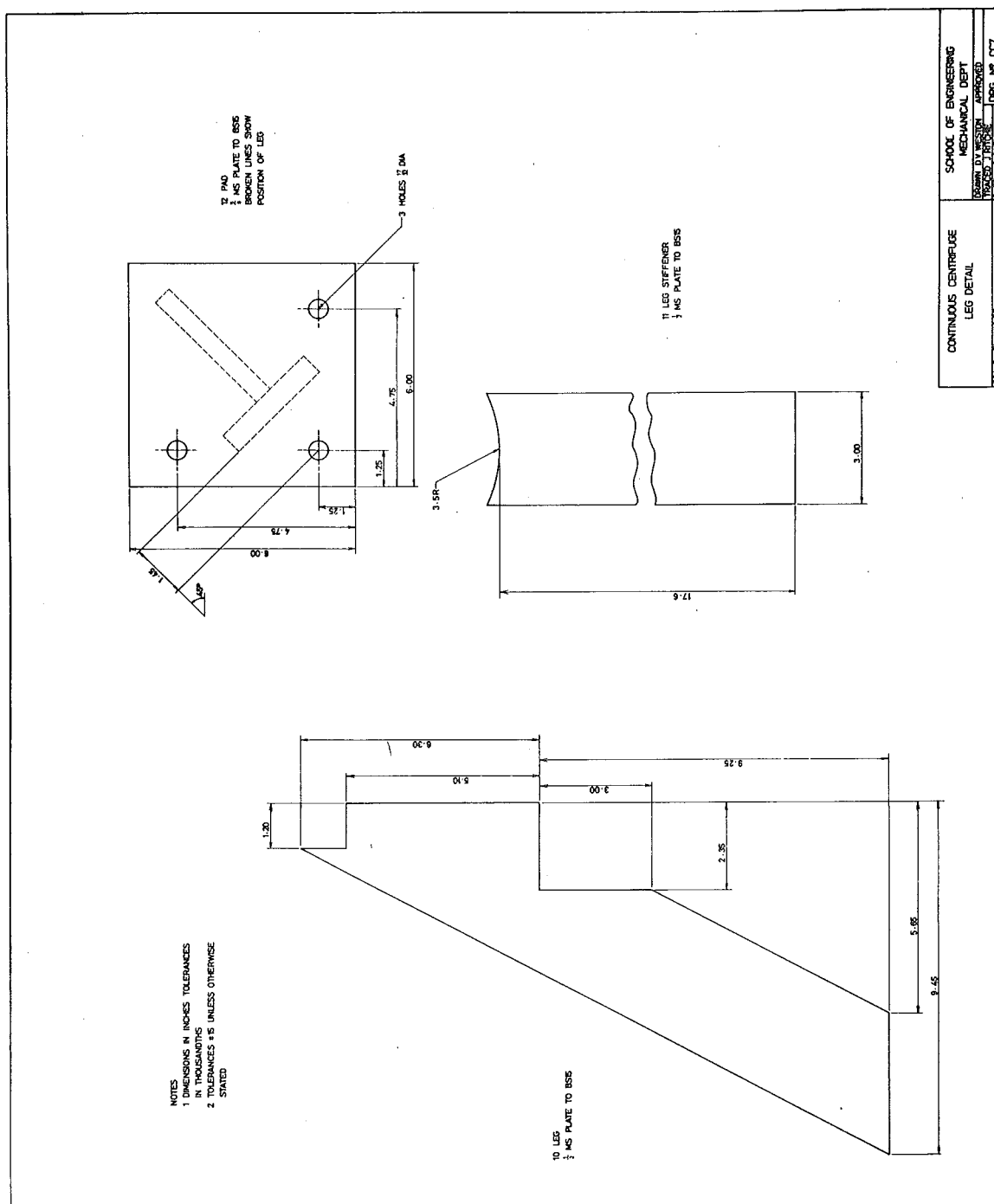
CC38 Modified spiral drum.

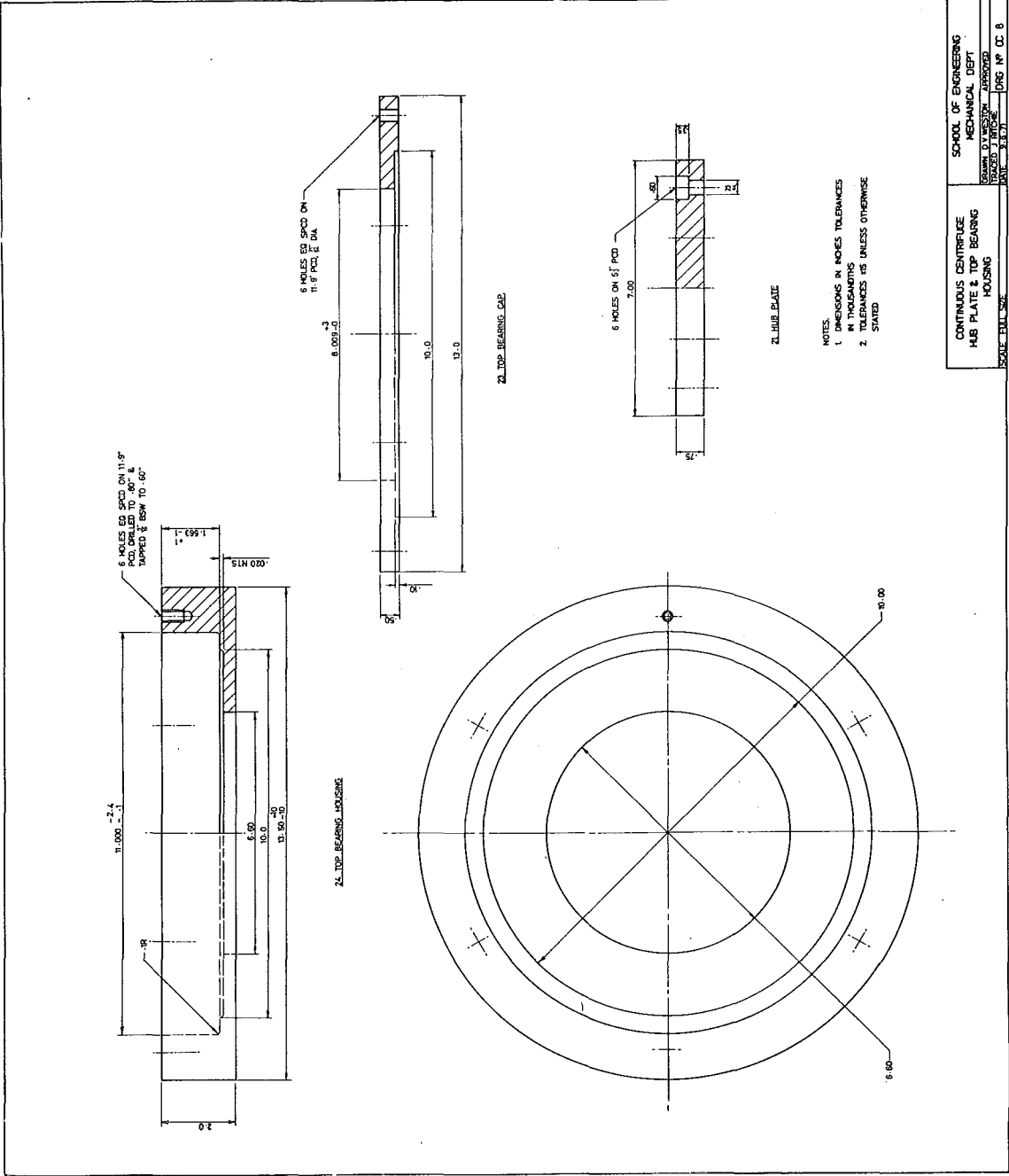




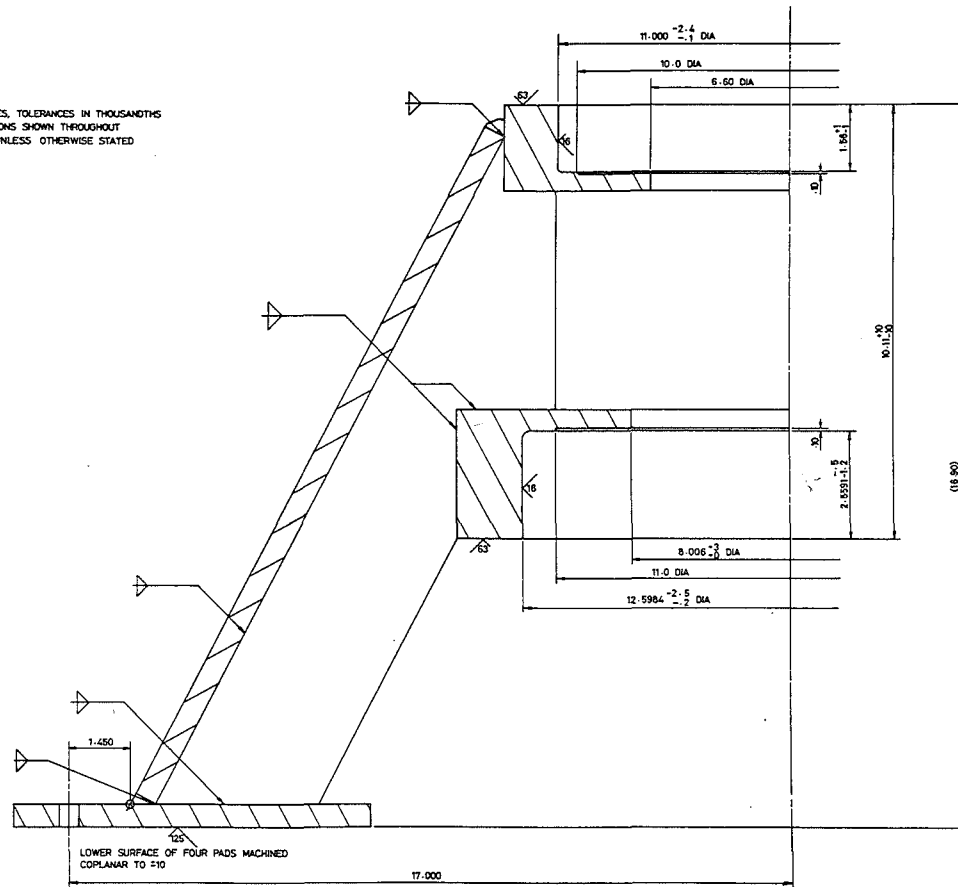


CONTINUOUS CENTRIFUGE BOTTOM BEARING HOUSING AND RETAINER RING		SCHOOL OF ENGINEERING MECHANICAL DEPT
DESIGNED BY: [blank]		DATE: 4.8.71
CHECKED BY: [blank]		DATE: [blank]
SCALE: FULL SIZE		DRG NO: 006

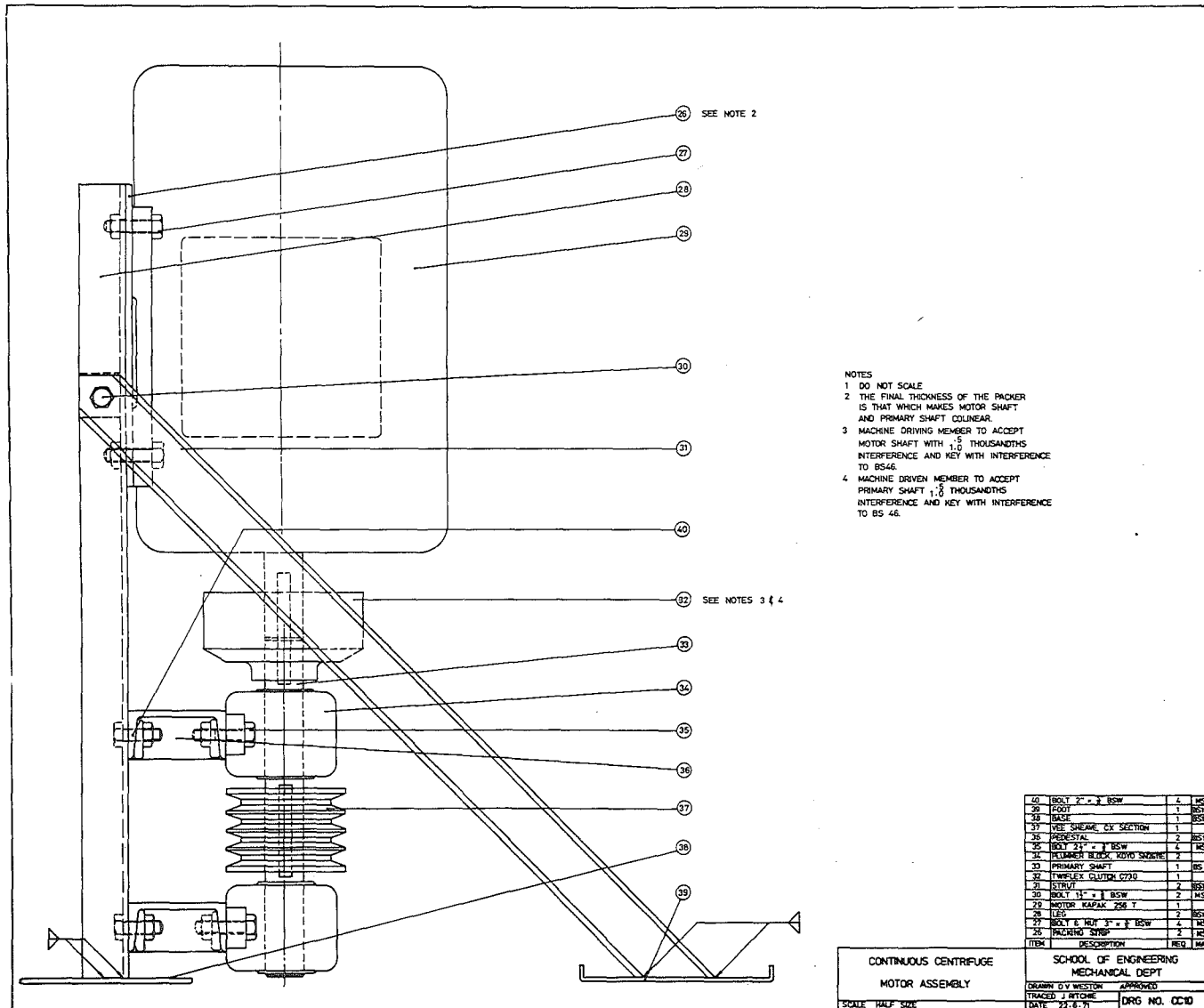




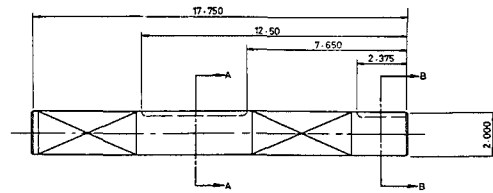
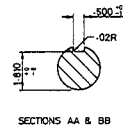
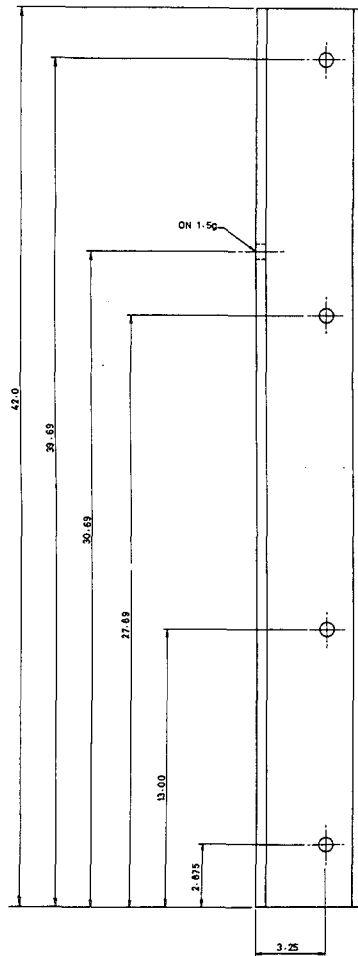
- NOTES
 1 DIMENSIONS INCHES, TOLERANCES IN THOUSANDTHS
 2 FINISHED DIMENSIONS SHOWN THROUGHOUT
 3 TOLERANCE $\pm .015$ UNLESS OTHERWISE STATED
 4 DO NOT SCALE



CONTINUOUS CENTRIFUGE		SCHOOL OF ENGINEERING	
MAIN BEARING & LEG ASSEMBLY		MECHANICAL DEPT	
DRAWN BY: J. H. H. H.		CHECKED BY: J. H. H. H.	
SCALE: FULL SIZE		DATE: 11.6.71	
		DRG NO: CC 9	



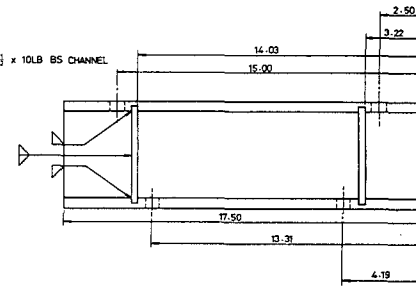
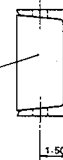
28. LEG
FROM 5 x 2½ x 10LB BS CHANNEL
TWO OFF, ONE LH, ONE RH



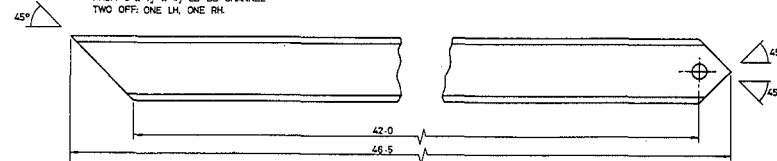
33. PRIMARY SHAFT
FROM .2" DIA BRIGHT SHAFTING

WEB STIFFENER
FROM ½ MS PLATE TO BS15

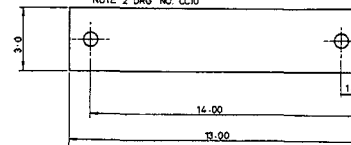
36. PEDESTAL
FROM 5 x 2½ x 10LB BS CHANNEL



31. STRUT
FROM 3 x 1½ x 4½ LB BS CHANNEL
TWO OFF, ONE LH, ONE RH

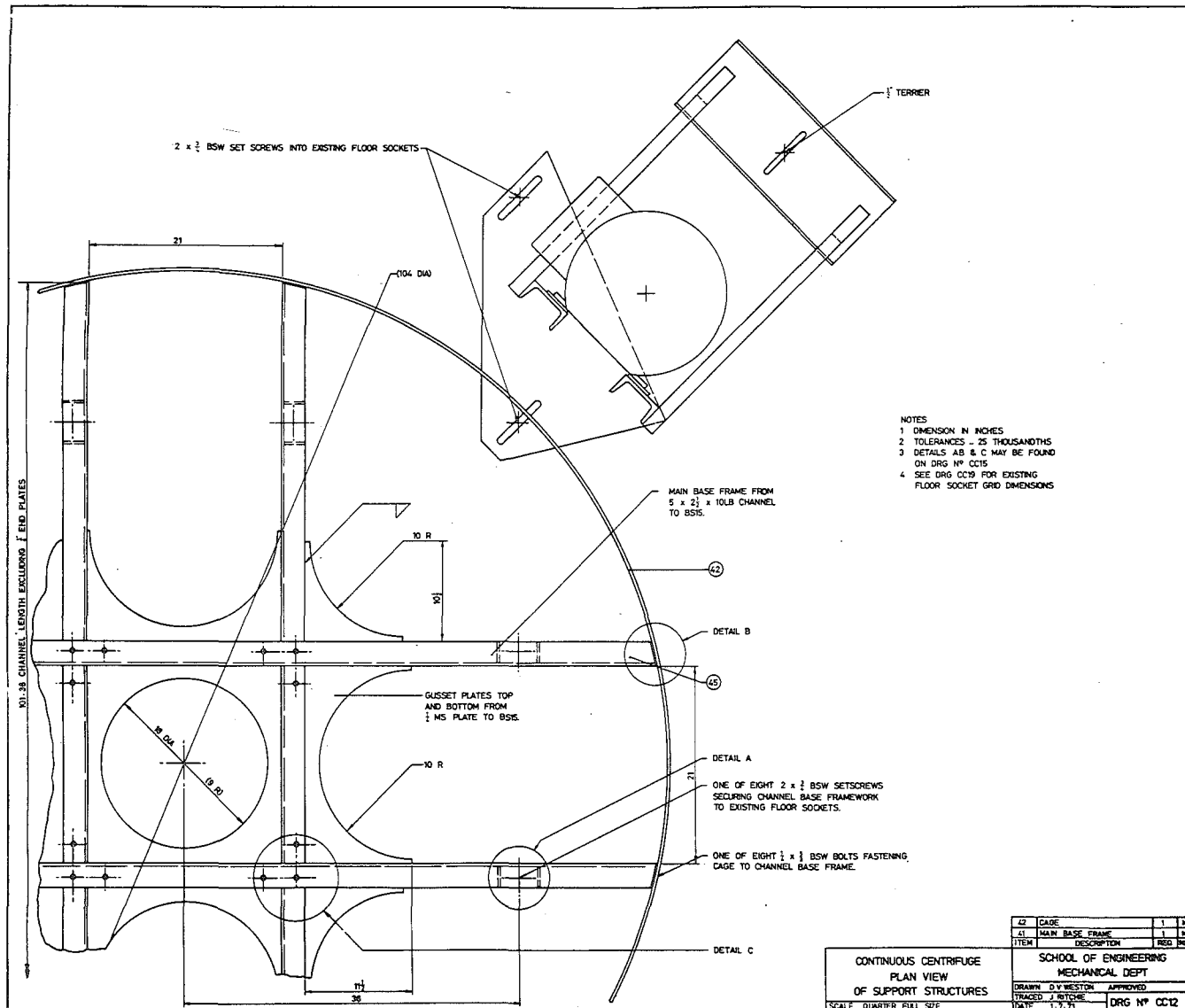


26. PACKING STRIP
FROM ½ STOCK M/C'D AS DESCRIBED
NOTE 2 DRG NO. CCT1



- NOTES
1. DIMENSIONS IN INCHES, TOLERANCES IN THOUSANDTHS
 2. TOLERANCES ± .15 UNLESS OTHERWISE STATED
 3. ALL HOLES DRILLED THRU ½" DIA
 4. TOLERANCE ON ANGLES ± 3°

CONTINUOUS CENTRIFUGE	SCHOOL OF ENGINEERING
MOTOR ASSEMBLY PARTS	MECHANICAL DEPT
SCALE: HALF SIZE	DRAWN: D.V. WESTON APPROVED
	TRACED: J. RITCHIE
	DATE: 12.6.71
	DRG N° CCT1



NOTE:
1 DIMENSIONS IN INCHES. TOLERANCES $\pm .025$
2 ANGULAR TOLERANCE $\pm 2^\circ$ UNLESS OTHERWISE STATED

38. MOTOR BASE PLATE FROM $\frac{1}{2}$ M.S. PLATE TO BSIS
WITH $\frac{1}{4}$ PACKING (AS SHOWN) FROM SIMILAR
MATERIAL (TEMPLATE SUPPLIED)

39. FOOT WITH $\frac{1}{4}$ PACKING STRIPS (AS SHOWN)
BOTH FROM $\frac{1}{2}$ MS PLATE TO BSIS.

PACKING STRIPS IDENTICAL

CONTINUOUS CENTRIFUGE
MOTOR SUPPORT BASE PLATES

SCHOOL OF ENGINEERING
MECHANICAL DEPT

DRAWN: D.V. WESTON
CHECKED: J. HITCHCOCK
DATE: 10.7.71

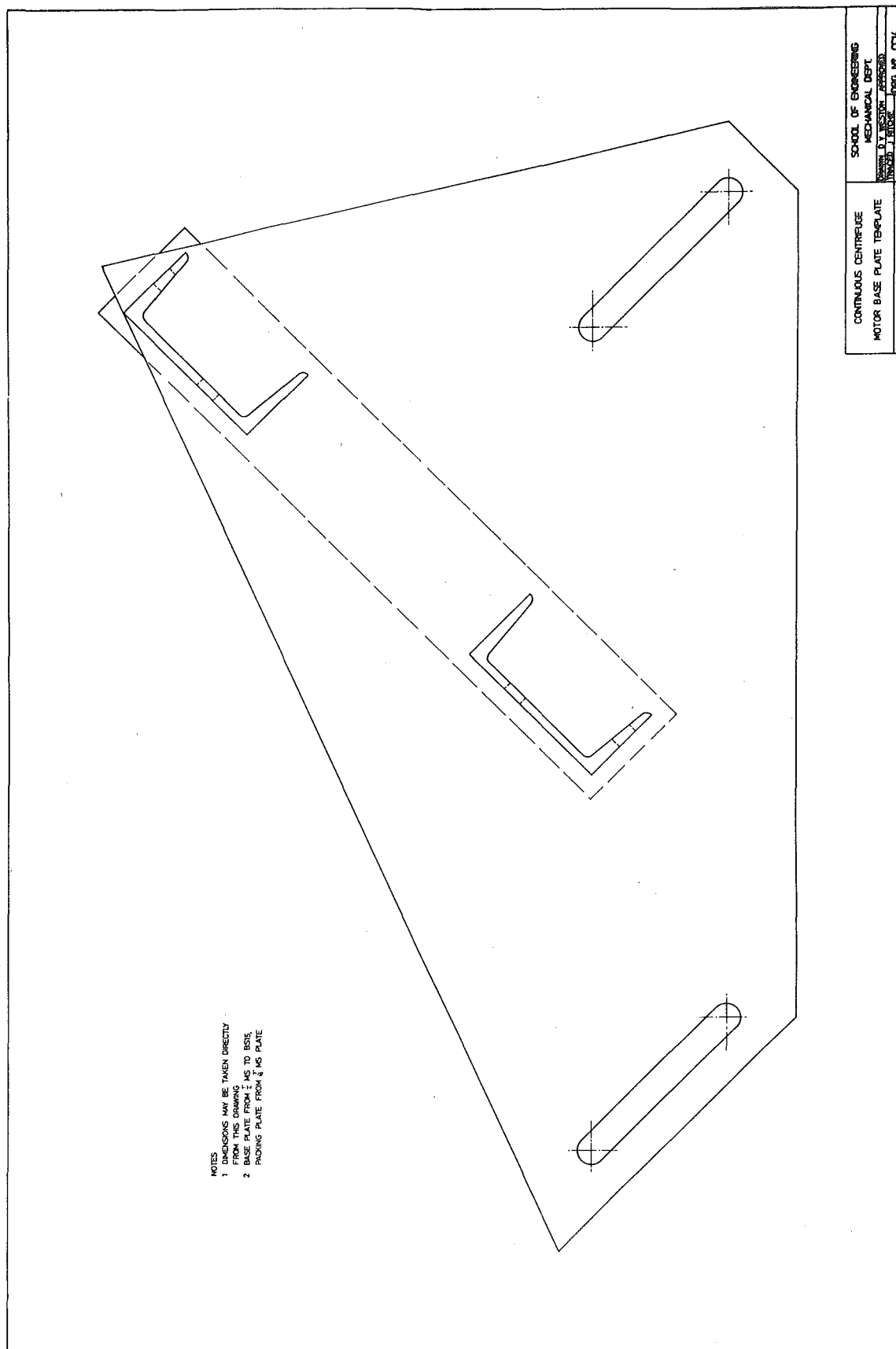
APPROVED:
DRG N° CC 13

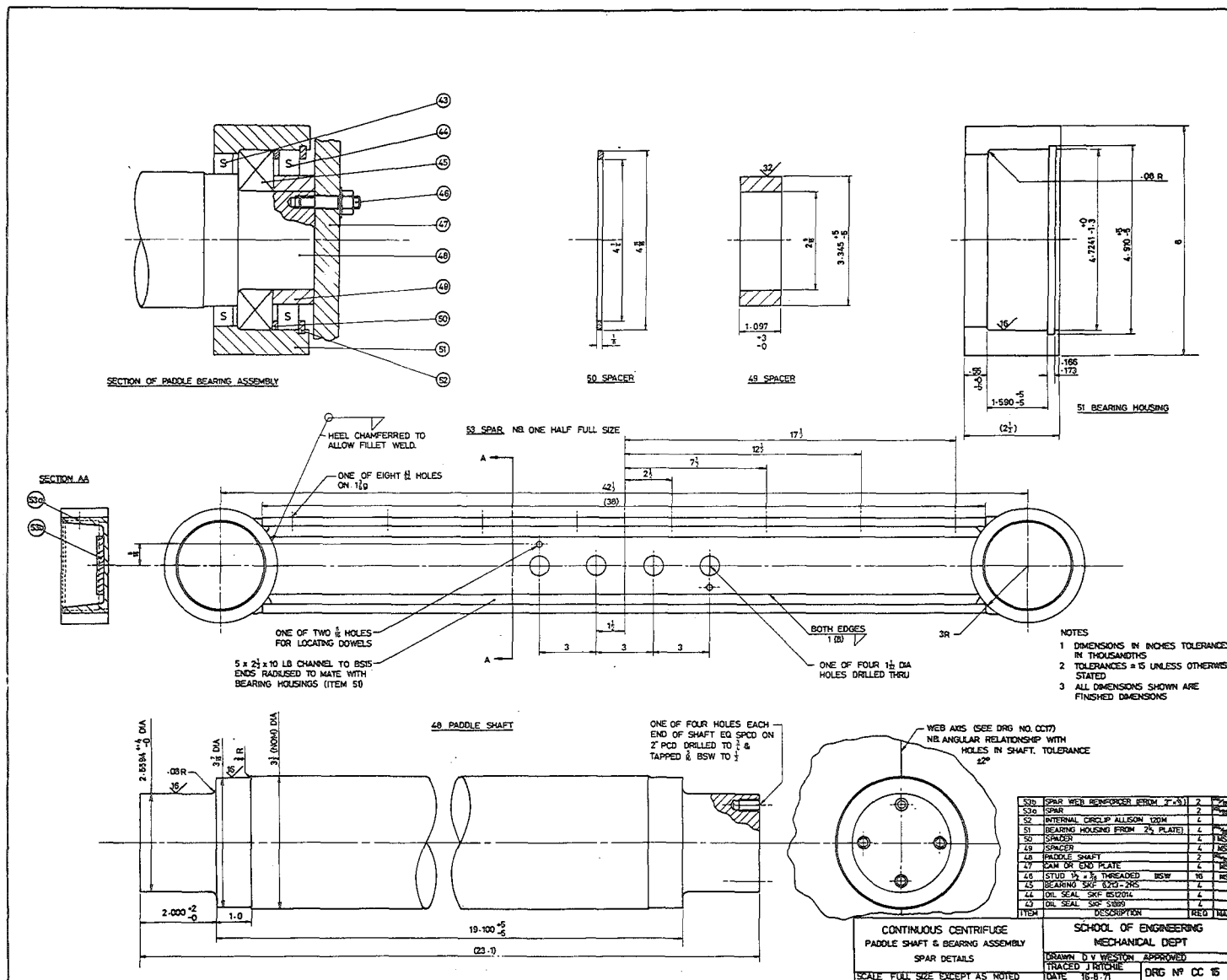
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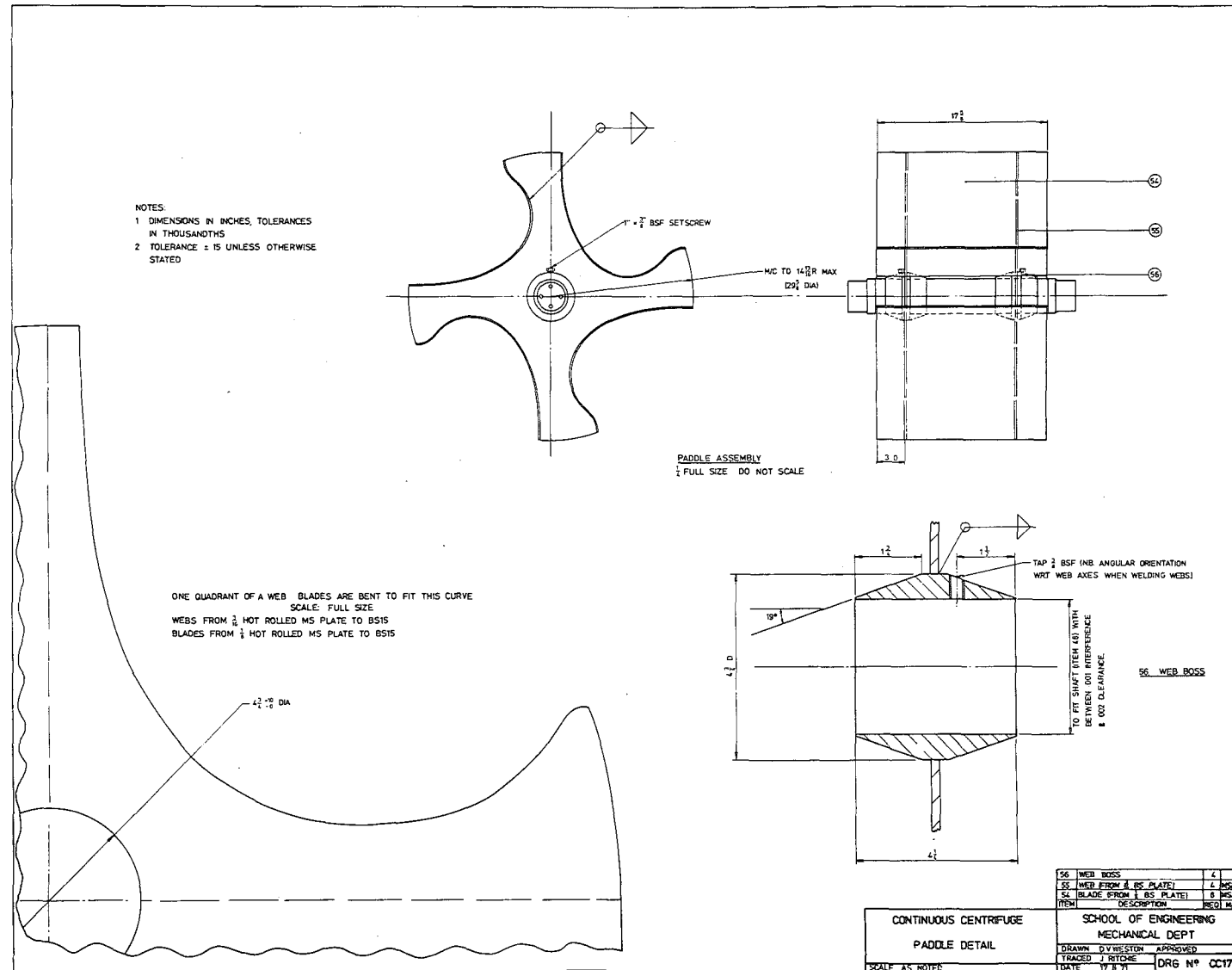
PACKING STRIPS IDENTICAL

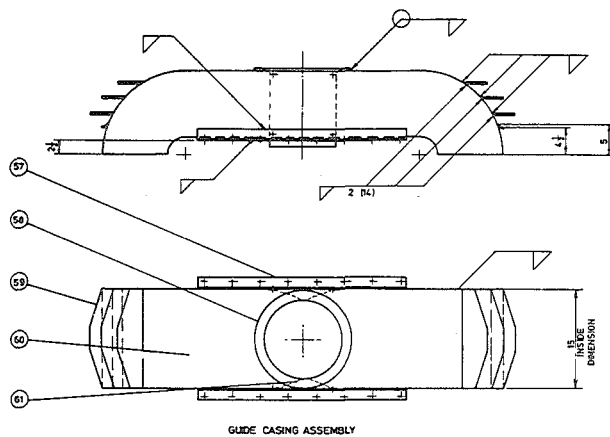
Technical drawing of a mechanical part, likely a bracket or support, showing two views: a front view (top) and a side view (bottom). The front view shows a rectangular base with a vertical support on the right. The side view shows the profile of the part with dimensions. Key dimensions include: overall width 9, overall height 19, base thickness 2.5, support thickness 2.4, and various internal dimensions like 5.3, 5.5, 1.75, 5.62, and 12. The drawing includes dashed lines for hidden edges and a section line A-A.

CONTINUOUS CENTRIFUGE MOTOR SUPPORT BASE PLATES	SCHOOL OF ENGINEERING MECHANICAL DEPT	
	DRAWN D V WESTON THROCK J RITCHIE	APPROVED DRG N° CC13
SCALE HALF SIZE	DATE 10.7.71	

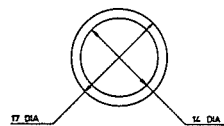




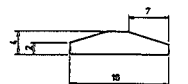




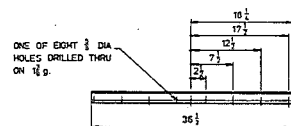
GUIDE CASING ASSEMBLY



58 ENTRY RING, FROM 1/8" PLATE

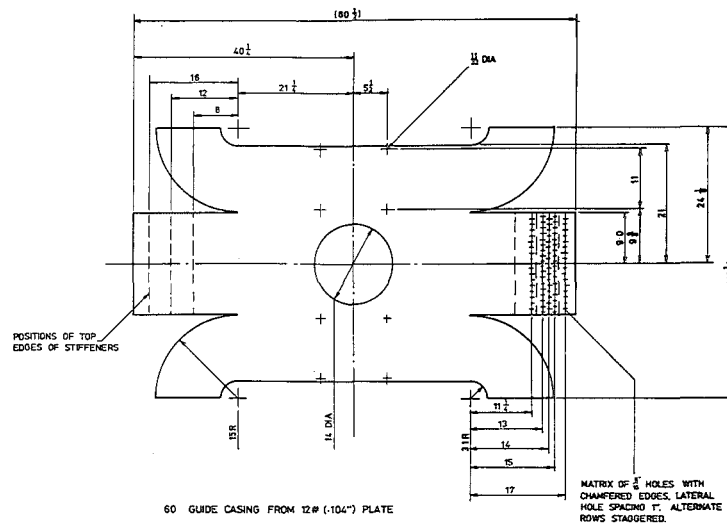


59 STIFFENERS FROM 1/8" PLATE

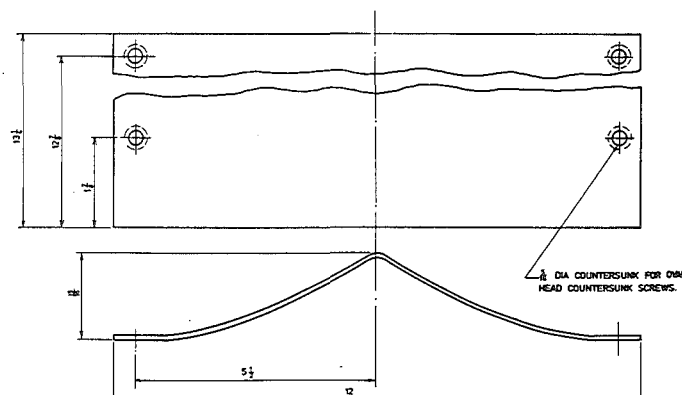


57 GUIDE CASING MOUNT FROM 2 x 2 x 1/4" ANGLE

NOTES:
1 DIMENSIONS IN INCHES
2 TOLERANCES ± .01



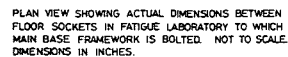
60 GUIDE CASING FROM 12# (1-104") PLATE



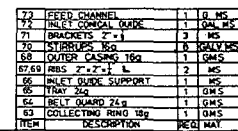
61 GUIDE PLATES, FROM 12# PLATE, FULL SIZE

ITEM	DESCRIPTION	QTY
62	1/2" DIA. COUNTERSUNK SCREWS	8
61	GUIDE PLATES	2
60	GUIDE CASING	1
59	GUIDE CASING STIFFENERS	6
58	ENTRY RING	1
57	GUIDE CASING MOUNT	2

CONTINUOUS CENTRIFUGE GUIDE CASING ASSEMBLY		SCHOOL OF ENGINEERING MECHANICAL DEPT
SCALE: ONE FULL SIZE OR AS NOTED	DATE: 7/28/77	DRG NO: CC 18

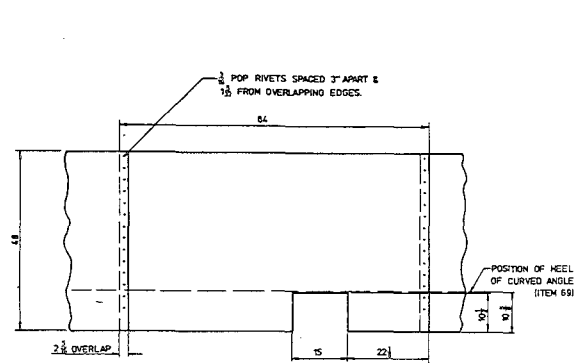


- NOTES:
1. OUTLET CASING IS FROM FOUR SHEETS OF 7' x 4' 16 G GALV MS SHEET, SINGLE LAP JOINED WITH A SINGLE ROW OF $\frac{3}{4}$ " POP RIVETS, 3" SPACING, COINCIDENT WITH ALTERNATE MAIN BASE FRAME SUPPORT POINTS AS SHOWN. SEE DRG. CC 20 FOR BELT CUTAWAY DETAILS AND OVERLAP. N.B. DIRECTION OF OVERLAP.
2. SEE DRG. CC 20 FOR THIS DETAIL. RIVET SPACING, 9".
3. COLLECTING RING MUST HAVE OVERLAPS IN DIRECTION SHOWN.
4. THESE CONNECTIONS BETWEEN OUTLET CASING AND 2" x 2" x $\frac{1}{2}$ " ANGLES ARE $\frac{3}{4}$ " POP RIVETS AT 4" SPACING (SINGLE ROW).
5. CONNECTION BETWEEN INLET GUIDE SUPPORT STRUCTURE (ITEM 66) AND TOP CURVED ANGLE (ITEM 67) IS ONE $\frac{3}{4}$ " BSW x 1" BOLT AND NUT.
6. BELT BRACKET BRACKETED TO INLET GUIDE SUPPORT STRUCTURE IS A ROW OF $\frac{3}{4}$ " POP RIVETS SPACED 7" (ON VERTICAL SECTIONS OF ROW ONLY).
7. RT ANGLE BRACKETS FROM 2" x $\frac{1}{2}$ " SECURED TO OUTLET CASING WITH $\frac{3}{4}$ " POP RIVETS AND TO FLOOR SOCKETS WITH 7 BSW SCREW. (8 BRACKETS ONLY)
8. STIRRUPS (ITEM 70) SECURED TO COLLECTING RING ONLY WITH $\frac{3}{4}$ " POP RIVETS.
9. ONE $\frac{3}{4}$ " POP RIVET EACH SIDE.
10. CONICAL GUIDE AND FEED CHANNEL SIDEPLATES, POP RIVETED TO INLET GUIDE SUPPORT.
11. SIDEPLATES RIVETED TO INLET GUIDE SUPPORT.
12. DO NOT SCALE.

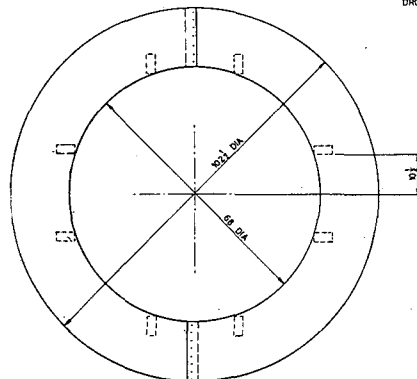


SCALE 1" = 1'-0"

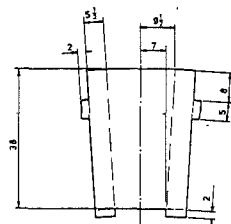
DRAWN D V WESTON		APPROVED
TRACED J RITCHIE		DRG N° CC 19
DATE 6-11-71		



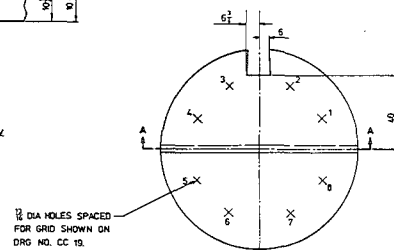
68. SINGLE SECTION OF OUTER CASING (DEVELOPMENT) HAVING CUTOUT FOR BELTS. OTHER THREE SECTIONS HAVE SIMILAR DIMENSIONS BUT WITHOUT CUTOUT. FROM 16g GALV. MS. PLATE. VIEW FROM INSIDE. FINISHED I.D. IS 104"



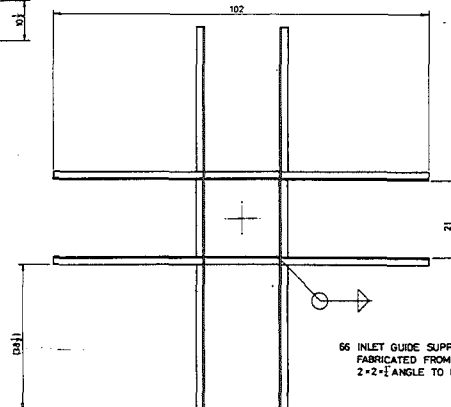
63. COLLECTING RING (PLAN VIEW) FROM 16g GALV. MS. THIS PART MAY BE MADE FROM TWO OR MORE SECTIONS PROVIDED DIRECTION OF OVERLAP IS AS SHOWN.



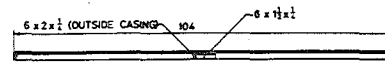
64. DEVELOPED VIEW OF BELT GUARD, FROM 24g GALV. MS. DASHED LINES INDICATE FOLDS.



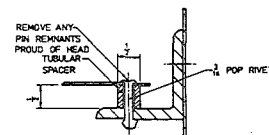
65. TRAY FROM TWO OR MORE SHEETS OF 24swg GALV. MS. LAP JOINTED & SOLDERED ON SITE. NB. 1:24 SCALE



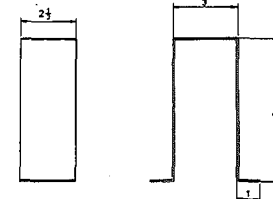
66. INLET GUIDE SUPPORT, FABRICATED FROM 2x2x1/4 ANGLE TO BS15.



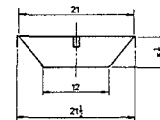
67, 69. RIBS, EACH IN TWO SECTIONS OF 2x2x1/4 MS TO BS15, JOINED AS SHOWN, (NB. LOWER RIB HAS TOE FACING UPWARDS).



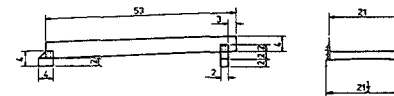
DETAIL OF JOINT BETWEEN COLLECTING RING AND LOWER RIB (NOTE 2, DRG CC 19)



70. STIRRUP, FROM 16g GALV. MS. LENGTH BEFORE BENDING, 17 1/2" (SHOWN INVERTED NB. HALF FULL SIZE)



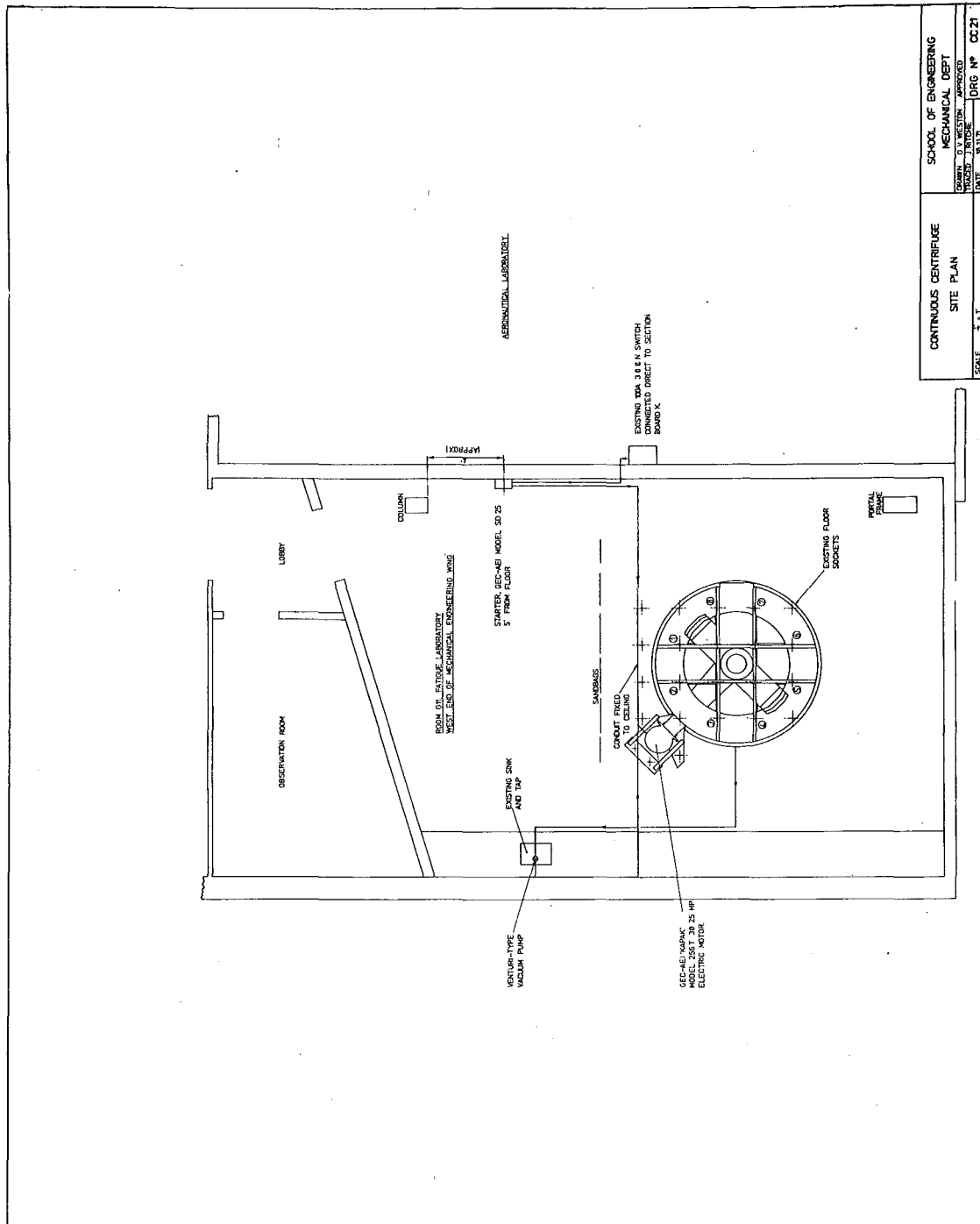
72. INLET CONICAL GUIDE, ELEVATION (LEFT) AND DEVELOPMENT (RIGHT) FROM 24g GALV. MS. NB. SCALE-ONE EIGHTH FULL SIZE.

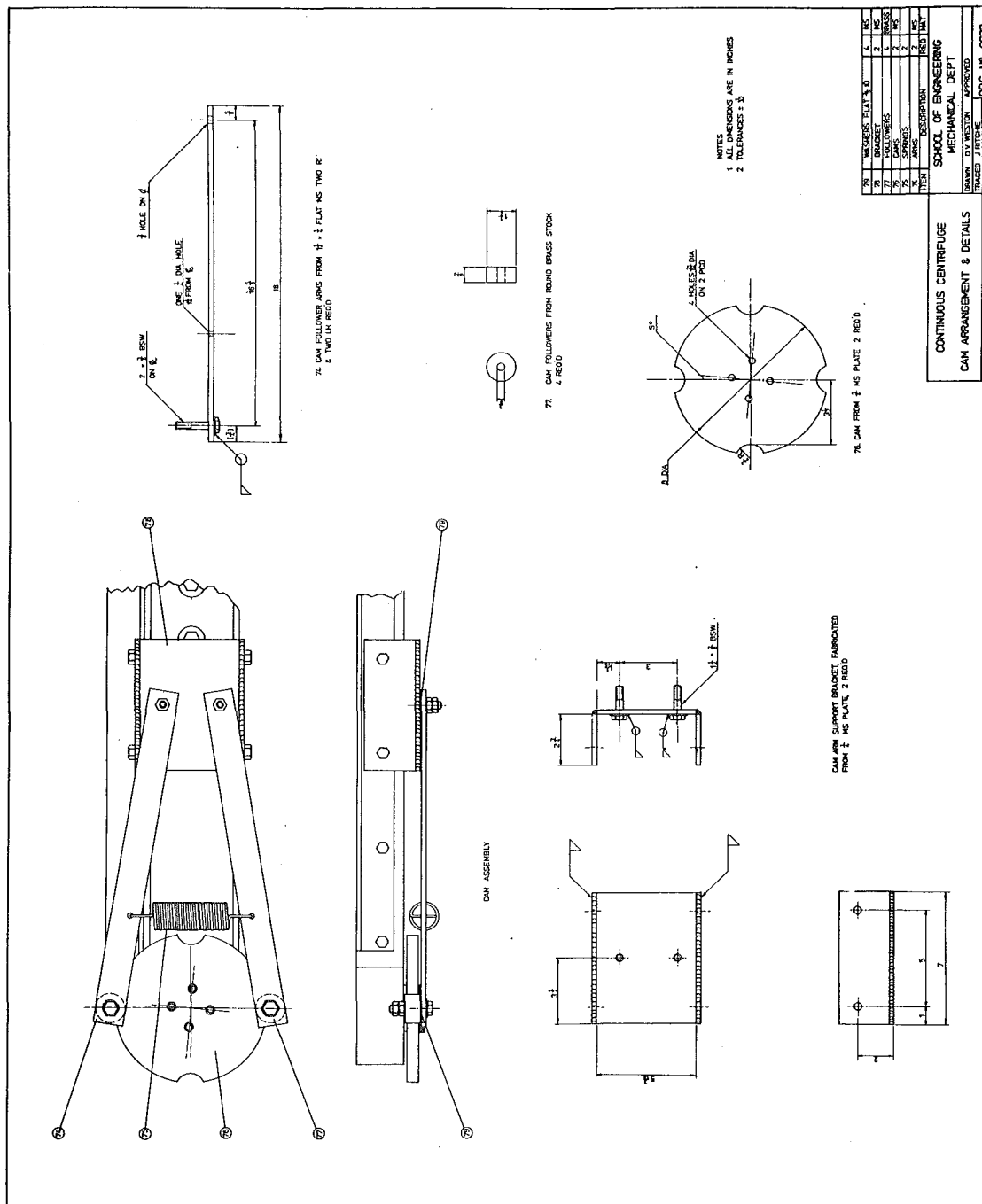


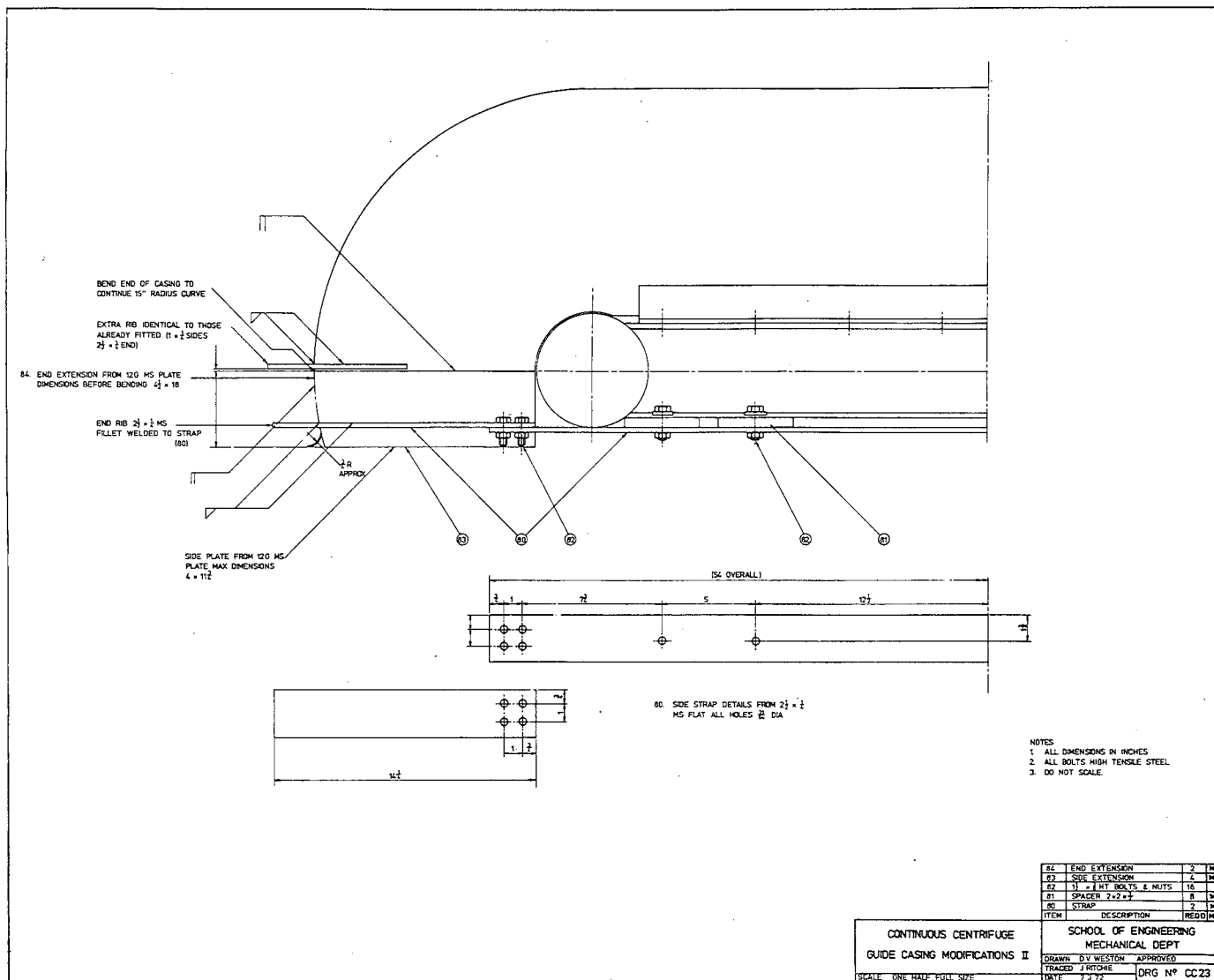
73. FEED CHANNEL FROM 24g GALV. MS. SIDE PLATES FROM 16g GALV. MS. TOLERANCES ± 1/8"

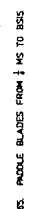
- NOTES:
1. DIMENSIONS ARE IN INCHES
 2. TOLERANCES ± 1/8" UNLESS OTHERWISE STATED.
 3. JOINT AND ASSEMBLY DETAILS SEE DRG CC 19
 4. DO NOT SCALE

CONTINUOUS CENTRIFUGE CASING AND SUPERSTRUCTURE DETAIL	SCHOOL OF ENGINEERING MECHANICAL DEPT
DRAWN: D.V. WESTON CHECKED: J. RITCHIE	APPROVED:
SCALE: 1" = 1'-0" OR AS INDICATED	DATE: 18.11.79
	DRG NO. CC20





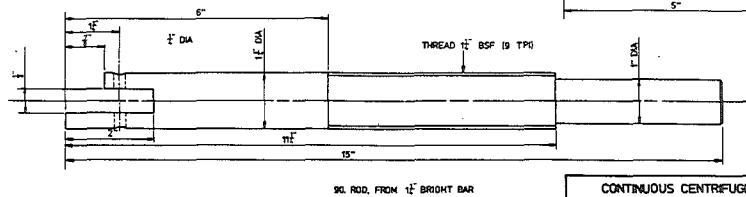
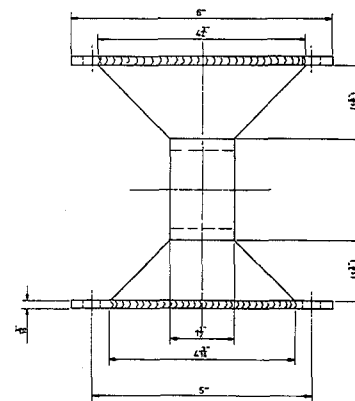
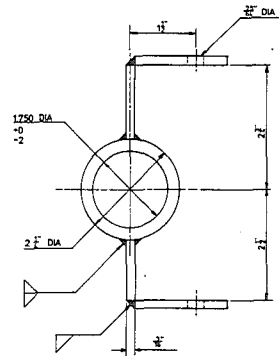
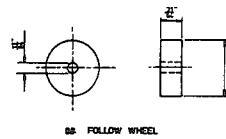
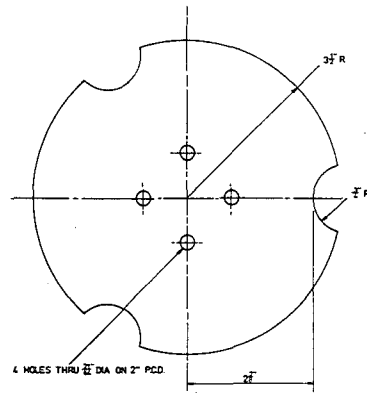
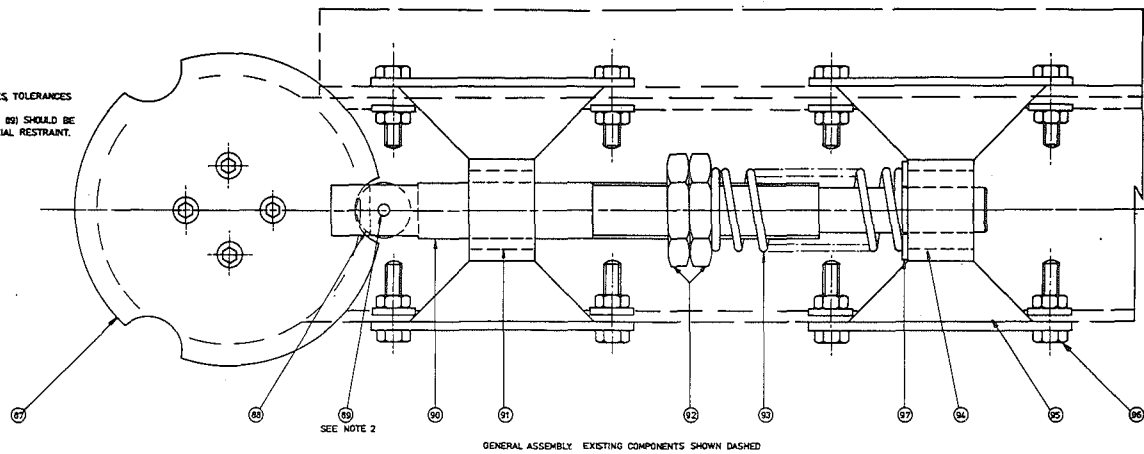




CONTINUOUS CENTRIFUGE		SCHOOL OF ENGINEERING	
MK II PADDOLE WHEEL DETAIL		MECHANICAL DEPT	
TRADED	3 V SYSTEM	APPROVED	
DATE	8-1-72	ORG NO.	CC 24
SCALE	1/2" = 1" FULL SIZE		

NOTES

1. DIMENSION IN INCHES, TOLERANCES IN THOUSANTHS.
2. ENDS OF PIN (ITEM 89) SHOULD BE PEENED TO GIVE AXIAL RESTRAINT.

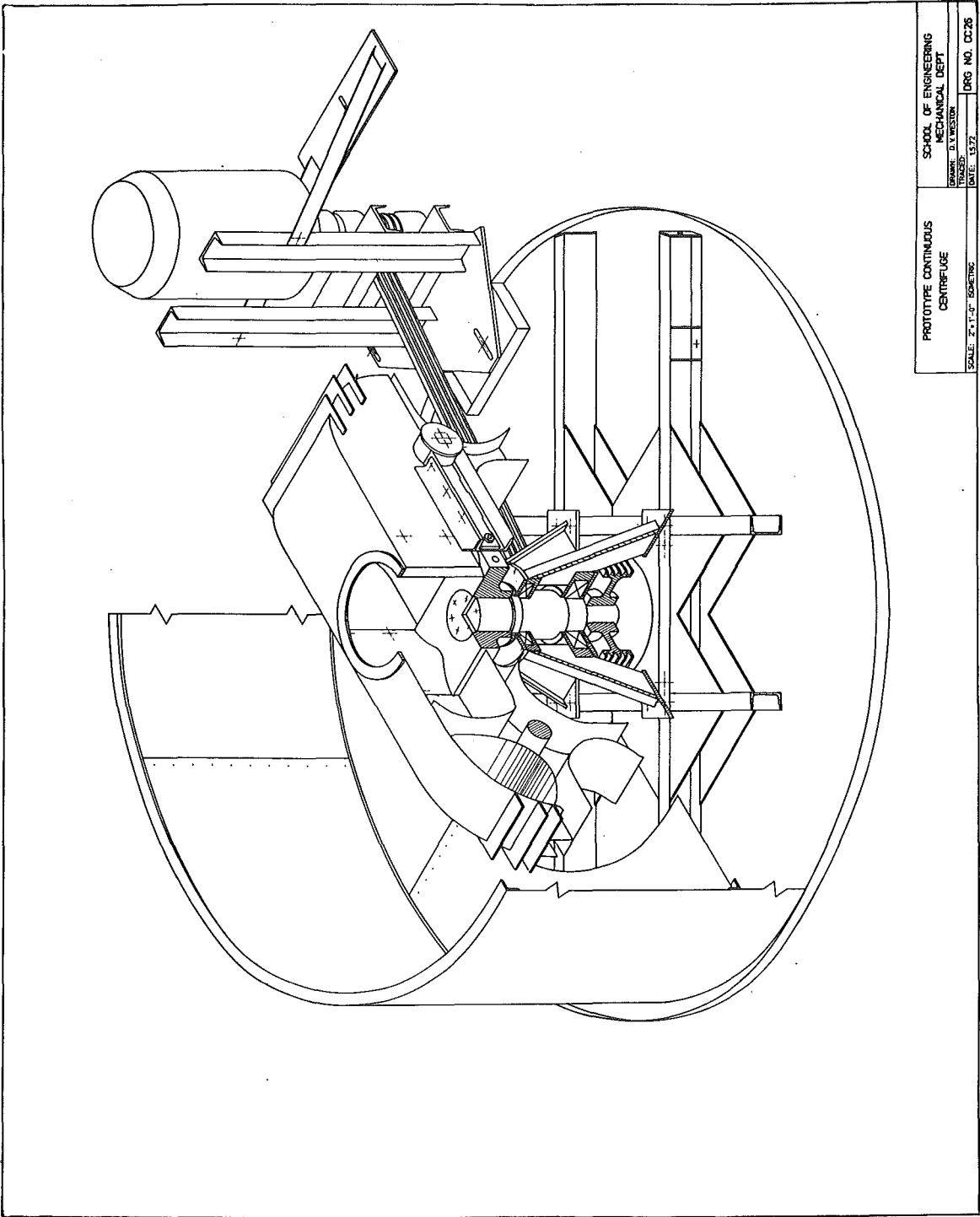


ITEM	DESCRIPTION	REMARKS
87	1/8 10 x 1/2 WASHER	2 MS
88	1/2 1/2 x 1/2 HT BOLTS	10
89	SUPPORTS	4 MS
90	BUSH	2 BRASS
91	SPRING	2
92	NUT 1/2 BSP	4 MS
93	BUSH	2 BRASS
94	ROD	2 MS
95	PIN	2 MS
96	FOLLOWER WHEEL	2 BRASS
97	CAM	2 MS

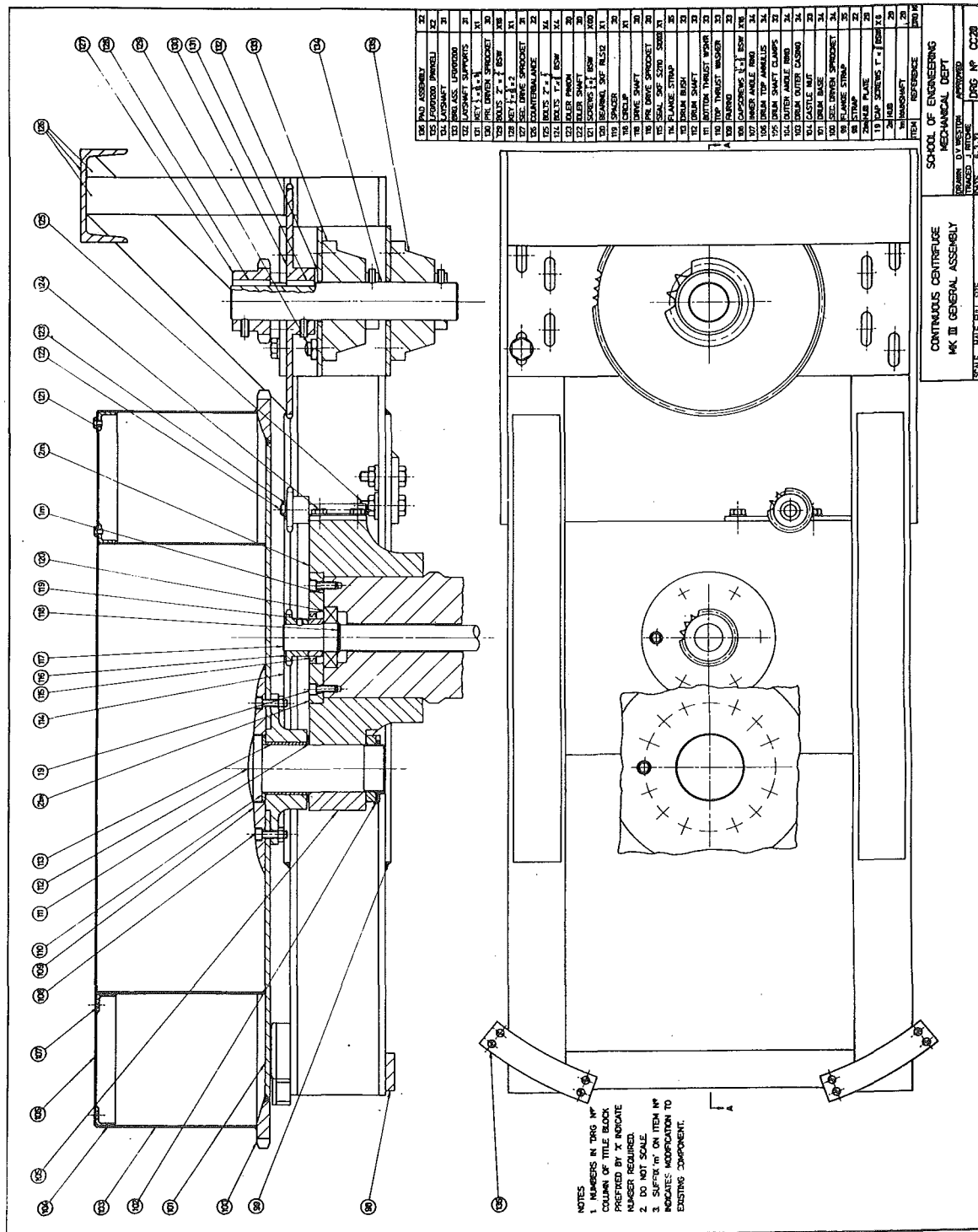
CONTINUOUS CENTRIFUGE
MK II CAM & FOLLOWER
SCALE: FULL SIZE

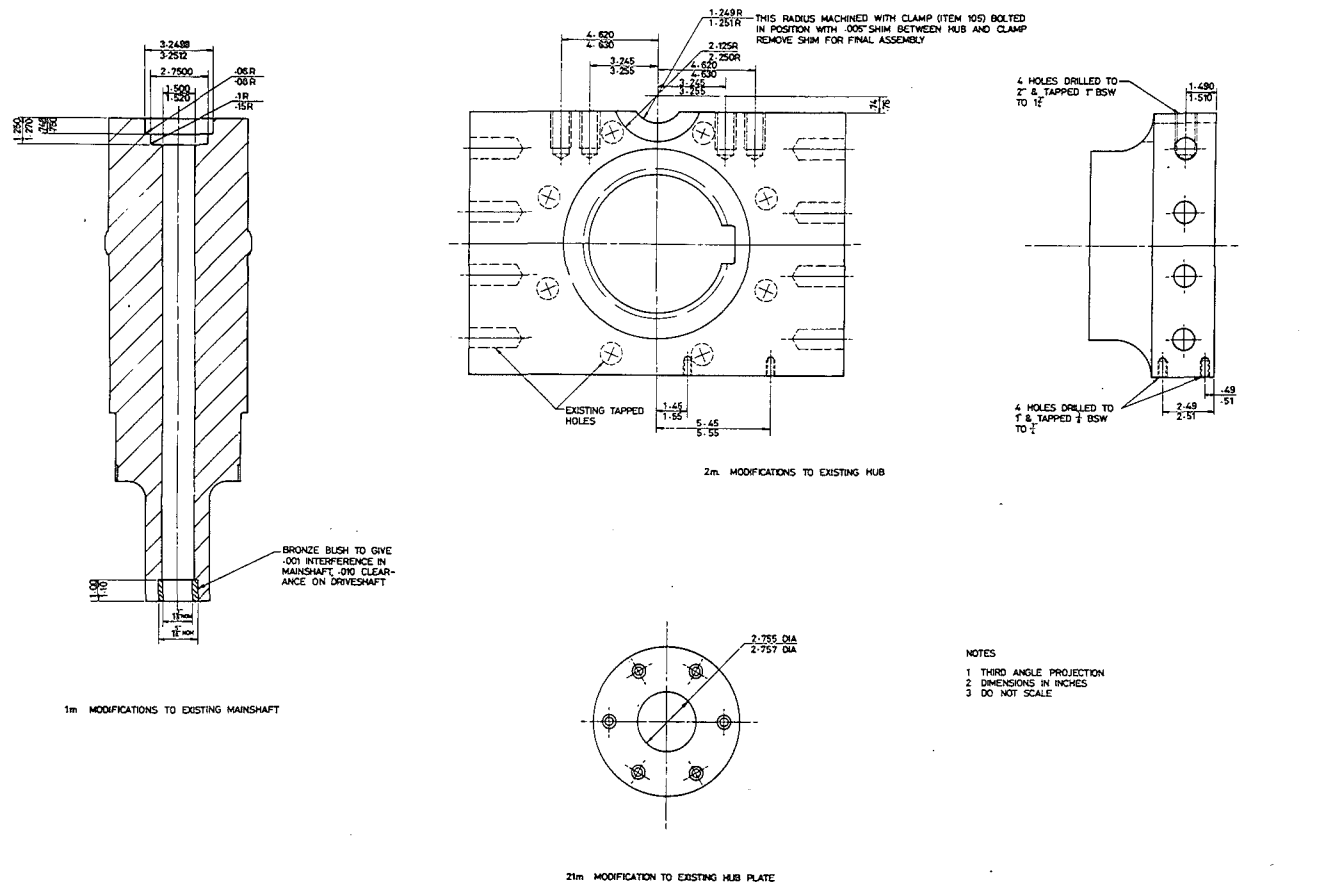
SCHOOL OF ENGINEERING
MECHANICAL DEPT.
DRAWN: D.Y. WESTON
TRACED: J. RITCHIE
DATE: 12.2.72

DRG N° CC25





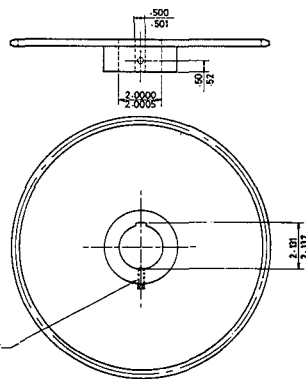
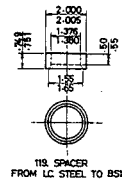
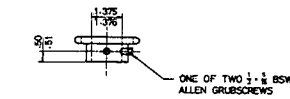
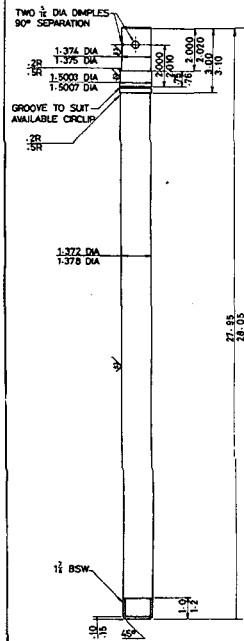


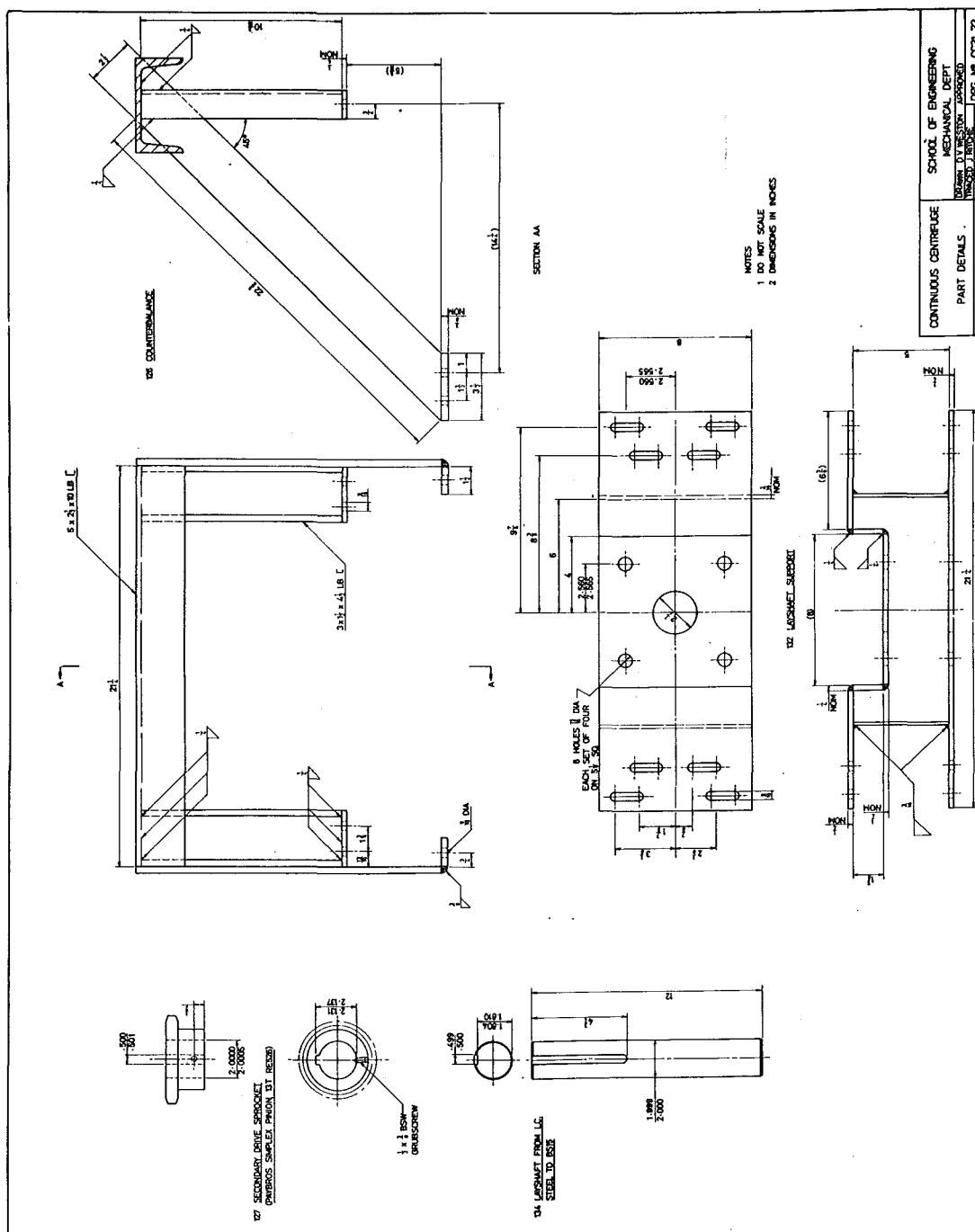


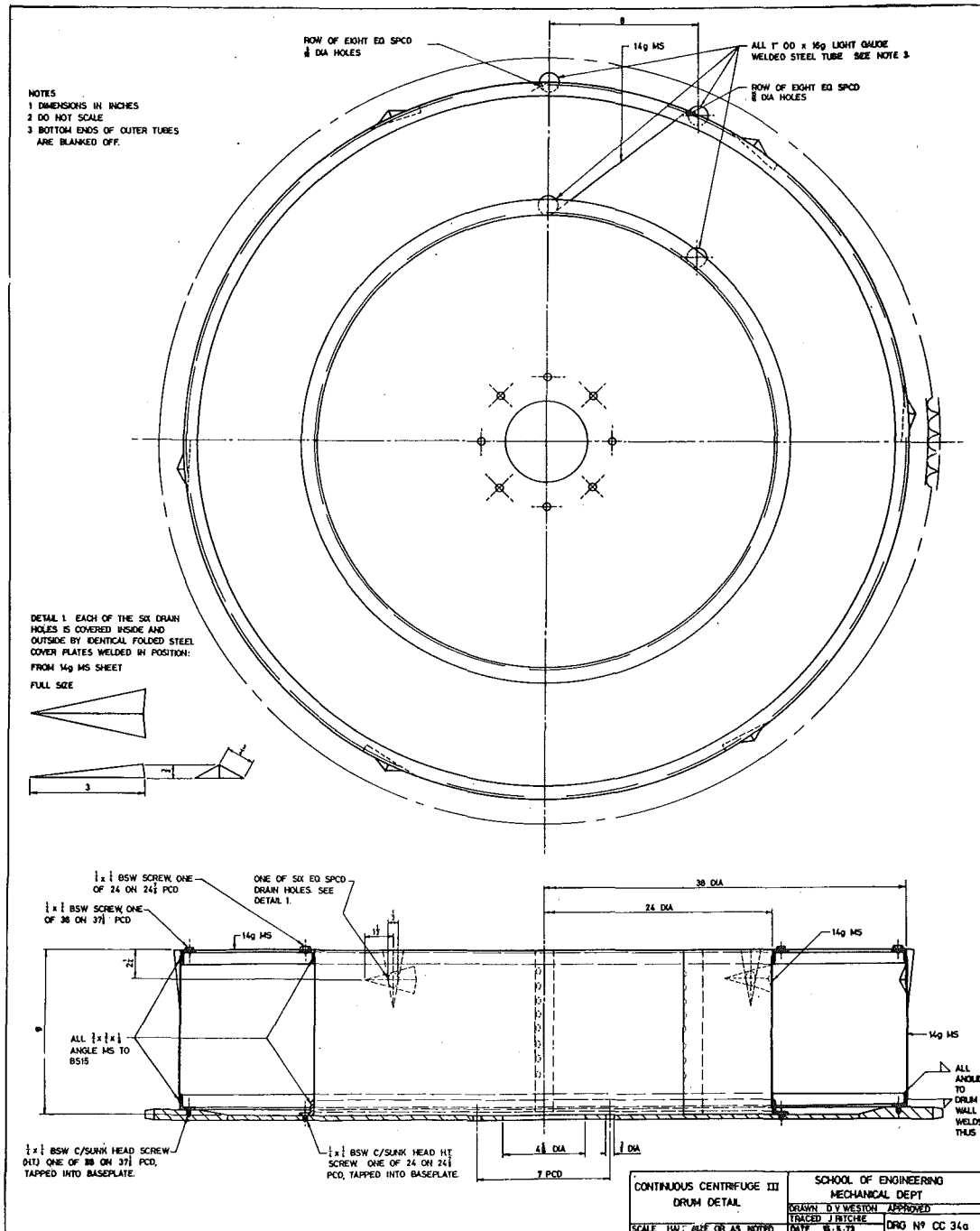
- NOTES
- 1 THIRD ANGLE PROJECTION
 - 2 DIMENSIONS IN INCHES
 - 3 DO NOT SCALE

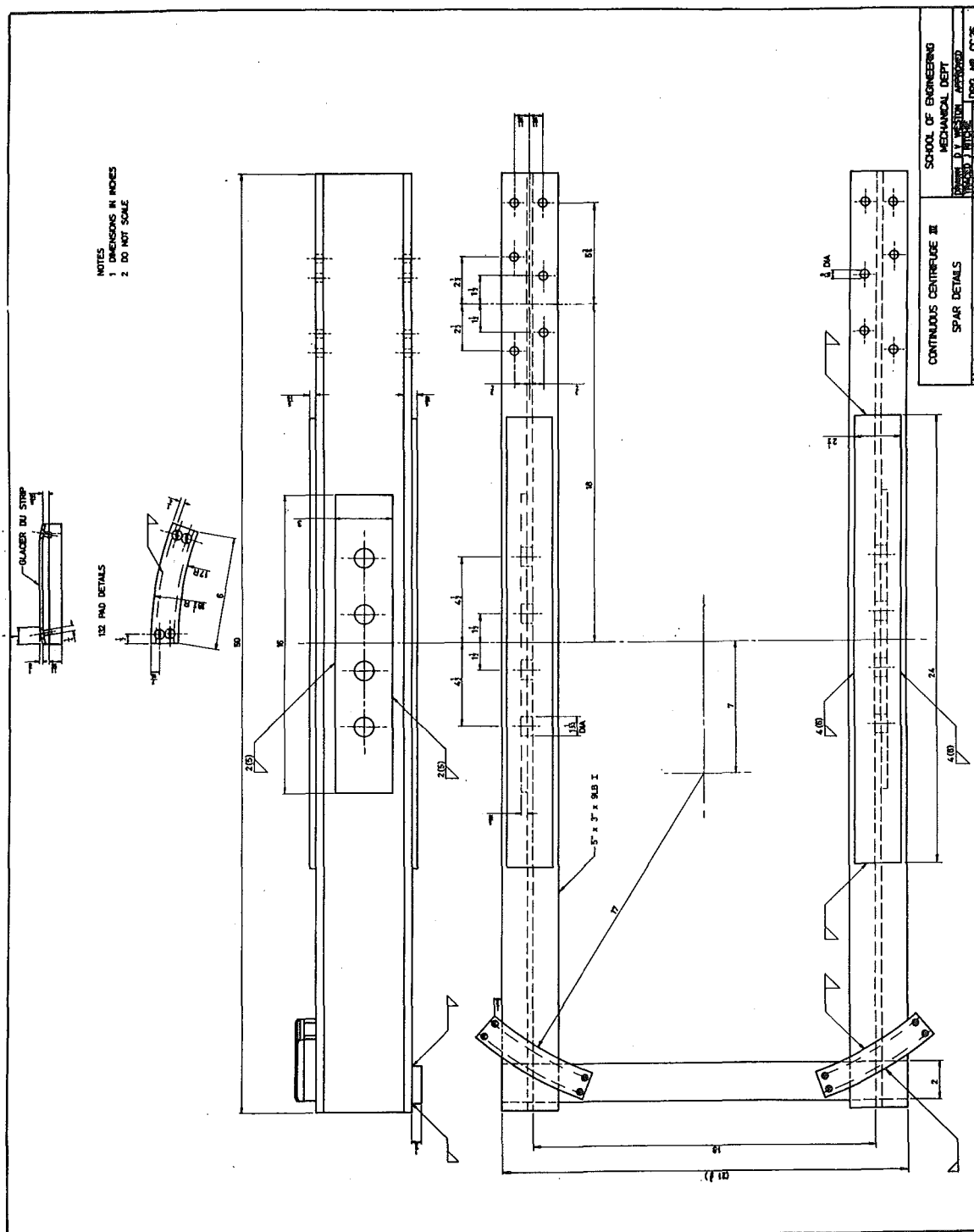
CONTINUOUS CENTRIFUGE COMPONENT MODIFICATIONS FOR MK III SYSTEM		SCHOOL OF ENGINEERING MECHANICAL DEPT	
DRAWN BY V WESTON APPROVED		TRACED J RITCHIE	
SCALE ONE HALF FULL SIZE		DATE 15.2.73	
		DRG NO CC29	

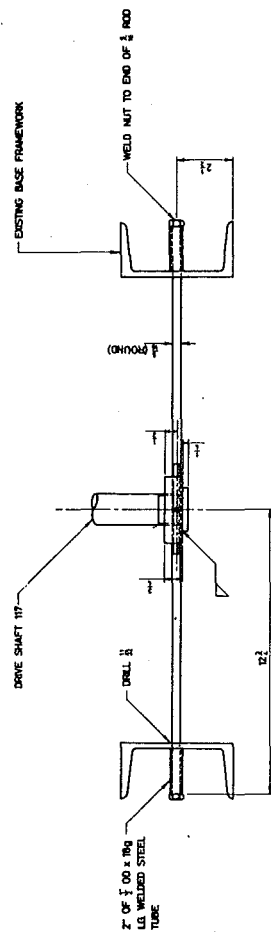
NOTES
1 ALL DIMENSIONS IN INCHES
2 DO NOT SCALE



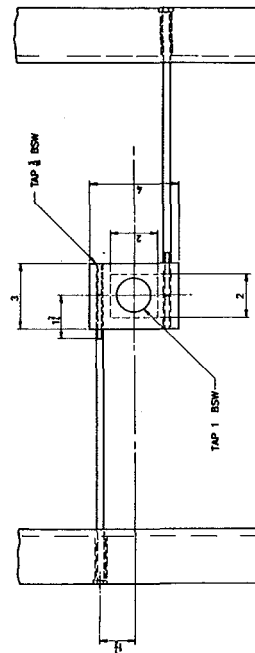








NOTES
 1 DIMENSIONS IN INCHES
 2 FIRST ANGLE PROJECTION
 3 DO NOT SCALE



108 DRIVE SHAFT STAY ASSEMBLY

CONTINUOUS CENTREFRUE II	SCHOOL OF ENGINEERING
DRIVE SHAFT STAY ASSEMBLY	MECHANICAL DEPT
SCALE: HALF SIZE	DESIGNED BY: M. S. SINGH
	DRAWN BY: M. S. SINGH
	CHECKED BY: M. S. SINGH
	DATE: 23.11.19
	ENG. NO. CC-26

